

OF THERMAL POWER STATIONS

M.I.Reznikov and Yu.M.Lipov

1. Steam Generation at Electric Power Stations 2. Power-producing Fuels and Their Characteristics 3. Fuel Preparation at Power Stations 4. Theoretical Principles of Combustion 5. Combustion Products 6. Efficiency of Fuel Heat Utilization 7. Pulverized Coal-fired Furnaces 8. Gas and Fuel Oil-fired Furnaces 9. Characteristics, Parameters and Motion Equations of Working Fluid 10. Temperature Conditions on Heating Surfaces 11. Hydrodynamics of Open Hydraulic Systems 12. Hydrodynamics of Closed Hydraulic Systems 13. Hydrodynamics of Bubbling Systems 14. Physico-chemical Principles of Behaviour of Impurities in Working Fluid 15. Water Conditions 16. Processes on the Fireside of Heating Surfaces 17. Evaporating Heating Surfaces 18. Steam Superheaters and Superheat Control 19. Low-temperature Heating Surfaces 20. Heat Exchange in Heating Surfaces of Boilers 21. Layout and Heat Calculation of Steam Boiler 22. Steam Boilers of High-capacity Monobloc Units 23. Steam Boiler Operation 24. Steam Generators of Nuclear Power Stations 25. Metals for Steam Boilers

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THERMAL POWER STATIONS

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PREFACE

This textbook has been written as a higher-education course in steam boilers for thermal power stations. It presents the theory of the processes which occur in steam boilers, designs of boilers for thermal power stations and steam generators for nuclear power stations, and the operating principles of boilers and steam generators.

The material in the book is based on four fundamental principles which are closely interrelated and reflect the current state of progress in science and technology: (1) the physico-chemical processes in the fuel, gas-air, and water-steam paths of modern highcapacity boilers; (2) the correlation between these physico-chemical processes and the design, layout and arrangement of steam boilers and their elements; (3) advanced technological processes and their technical and economical substantiation; and (4) the correlation between the processes occurring in boilers and the principles of boiler operation. This method of analysis encourages the optimal selection of technological processes, boiler designs, and operating regimes.

At the beginning of the course, we explain the role and place of the steam boiler in the general scheme of electric power production at modern high-capacity steam-turbine power stations, give the classification of steam boliers, describe the functions of the principal boiler elements and, in introductory form, the physico-chemical processes which occur in the water-steam, fuel and gas-air paths of boilers. Thus, the students are immediately introduced to the range of topics which are later discussed in more detail.

A number of chapters are devoted

to power-producing fuels and their characteristics, fuel preparation for combustion, the theoretical principles of combustion, technology of fuel combustion, and efficiency with which heat is utilized in steam boilers. Next the book focuses on the principles of hydrodynamics and the temperature and water conditions in steam boilers. This constitutes the range of problems related to the processes of steam generation.

Having studied the processes of fuel combustion and steam generation, the reader is acquainted with several particular designs of steam boilers and steam boiler elements. Special emphasis is placed on the processes and plants for high and supercritical steam parameters, monobloc units, the utilization of non-traditional fuels, and methods for increasing the reliability and efficiency of power plant equipment.

Furtheron, the book explains the principles, stages and sequence of heat and hydraulic calculations for steam boilers, including data on the application of electronic computers and the development of mathematical models of steam boilers. The concluding chapters are of a generalized nature and describe certain particular designs of modern steam boilers, trends in their development, and principles of boiler operation.

In view of recent progress and perspectives in nuclear power engineering and the construction of high-capacity nuclear power stations, of large theoretical and practical interest are data on the steam generators of nuclear power stations. For the first time in higher-education textbooks, some pro-

cesses occurring in the steam boilers of thermal power stations and steam generators of nuclear power stations are discussed in parallel. In addition, a separate chapter is devoted solely to the steam generators of nuclear

power stations.

The authors have carefully selected the illustrations for the book. For deeper analysis of the problems being studied, different types of boiler circuits and designs are compared in illustrations. In some illustrations, boilers or their elements are shown in a simplified form to facilitate the reader's understanding of how they function and the processes which take place in them.

The present book is the result of

many years of lecturing a course on steam generators of power stations at the Moscow power engineering institute, which has been initiated by Academician M. A. Styrikovich.

The authors would like to express special thanks to their colleagues on the faculty of steam generators of power stations at the Moscow power engineering institute [faculty chair Prof. V. S. Protopopov, Dr. Sc. (Eng.)]. the reviewers of the book, the faculty of steam generators at the Saratov polytechnical institute [faculty chair Prof. A. V. Zmachinsky, Dr. Sc. (Eng.)] and B. I. Shmukler, Cand. Sc. (Eng.), for their valuable comments on the manuscript.



STEAM GENERATION AT ELECTRIC POWER STATIONS

1.1. The Steam Boiler at a Power Station

An electric power station is an industrial plant for generation of electric energy. In the USSR and industrially developed countries, the major portion of electric energy is produced at fuelfired (thermal) power stations which utilize the chemical energy of combustion of organic fuels. A certain quantity of electricity is also produced at nuclear power stations, a kind of thermal stations which utilize the energy of nuclear fuels, and at hydraulic power stations which utilize the energy of falling water.

Irrespective of the type of station, electric energy is, as a rule, produced on a centralized basis, which means that individual power stations supply energy to a common power grid, and therefore, are combined into power systems which may cover a large territory with a large number of consumors. This principle increases the reliability of power supply to consumers, decreases the required reserve power, reduces the cost of produced energy due to more rational load on the power stations of a system, and allows the use of power plants of higher unit power. At some power stations, the centralized principle is employed for the supply of heat to consumers in the form of hot water and low-pressure steam, as well for the supply of electric energy. Electric power stations. electric and heat power networks and consumers make up what is called a power system. Individual power systems may be interconnected by hightension electric power lines into a power grid. Most of the power grids

in the Soviet Union comprise the supergrid, which is the highest form of organization of energy production.

Thermal power stations. Steam-turbine power stations are the main type of power stations operating on organic fuels. They are subdivided into condensation plants which produce electric energy only and heat-and-power plants which can produce both electric energy and heat.

Steam-turbine power plants are advantegeous over other types in that they permit concentration of an enormous power in a single unit, have a relatively high economic efficiency and require the lowest capital costs and short time of their construction. The main thermal units at a steam-turbine power station are a steam boiler and a steam turbine (Fig. 1.1). A steam boiler is a combination of heating surfaces in which steam is generated from continuously fed water by utilizing the heat liberated on combustion of organic fuel which is fed into the boiler furnace together with the air required for combustion. The water supplied into a steam boiler is called feed water. Feed water is preheated to the saturation temperature and vaporized and the saturated steam thus formed is further superheated.

As fuel is burned, it forms combustion products which sorve as a heattransfer agent in the heating surfaces where it gives up its heat to the water and steam which are called the working fluid. On passing the heating surfaces, the combustion products are cooled to a relatively low temperature and ejected from the boiler through a stack into the atmosphere. The stacks of high-power stations have a height

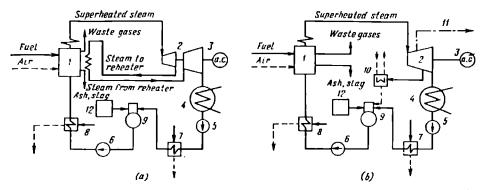


Fig. 1.1. Principal thermal diagram of (a) condensing station and (b) heat and power station t—steam boiler; 2—steam turbine; 3—clectric generator; 4—condenser; 5—condensate pump; 6—feed pump; 7—low-preasure heater; 8—high-pressure heater; 9—descript; 10—mains water heater; 11—industrial steam extraction; 72—water-treatment plant

of 200-300 m or even more to minimize local concentrations of contaminants in the air. Solid fuels leave ash and slag on combustion, which are disposed of from the boiler plant. The superheated steam produced in a boiler is supplied into a steam turbine where its thermal energy is converted into mechanical work on the turbine shaft. The latter is connected to an electric generator in which the mechanical energy is transformed into electricity. The waste, or dump, steam is fed from the turbine into a condenser, an apparatus in which the steam is cooled and condensed by means of cold water supplied from a natural (river, sea, pond) or artificial (cooling tower) water source.

power At modern condensation plants with a unit power of 150 MW or more, reheat superheating is employed, usually by arranging a single-bank reheat superheater (reheater) (Fig. 1.1a). Double-bank reheat superheaters are employed at power plants of a very high power; in this scheme, steam is returned to the boiler from two intermediate turbine stages. Reheat superheating increases the efficiency of a turbine and accordingly decreases the unit steam consumption for power generation; it also diminishes the moisture content of the steam in the low-pressure turbine stages and decreases erosion wear of turbine blades.

The condensate is pumped by a condensate pump through low-pressure water heaters into a deaerator, where the condensate is made to boil and is freed from oxygen and carbon dioxide that might cause corrosion of the equipment. Water from the deacrator is fed by means of a feed-water pump through a high-pressure water heater and then into the steam boiler. The condensate in low-pressure water heaters and the feed water in high-pressure water heaters are heated by the steam taken off from the turbine; this is called regenerative water heating. This method increases the efficiency of a steam-turbine plant and decreases the heat loss in the condenser.

Thus, the steam boiler of a condensation power plant (Fig. 1.1a) is supplied with the condensate formed from the steam produced in the unit. Part of this condensate is lost in the system as leakage. At heat and power stations, another portion of the steam produced is taken off and supplied as process steam to industrial consumers and for domestic purposes. At condensation plants, the steam leakage constitutes only a small fraction of the total steam consumption, around 0.5-1%, and is compensated for by make-up water pretreated in a watertreatment plant. At heat and power stations, the quantity of make-up

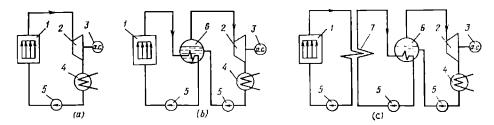


Fig. 1.2. Principal thermal diagrams of (a) single-circuit, (b) two-circuit and (c) three-circuit nuclear power stations

1-reactor; 2-steam turbine; 3-electric generator; 4-condenser; 5-pump; 6-steam generator; 7-intermediate heat exchanger

water added may attain 30-50% or even more.

Make-up water and turbine condensate contain certain impurities, mainly dissolved salts, metal oxides and gases. These impurities enter the steam boiler together with the feed water. In the course of vaporization, the concentration of impurities in the water increases and under certain conditions they can form deposits (scale) on the heating surfaces of the boiler, which impair heat transfer. Further, impurities of the water can partially pass to steam on vaporization, which should be avoided where possible, since the steam must be perfectly pure so as to prevent deposition of impurities in the turbine. For the two reasons mentioned, high contamination of feed water is inadmissible; the concentration of impurities in feed water is regulated by special standards and must be strictly controlled.

The operation of a steam boiler depends on a number of auxiliary devices and mechanisms, which include fuel-preparation devices, feed pumps to supply feed water to the boiler, forced-draft fans to supply air for combustion, induced-draft fans to eject combustion products through the stack into the atmosphere, etc. A steam boiler and the whole complex of its auxiliary equipment constitute what is called the boiler plant, or boiler installation. A modern boiler plant is a complicated engineering facility for steam generation in which all working

processes are mechanized and automatically controlled; an automatic protection system prevents failures of a boiler plant and enhances the reliability of its operation.

The principal trends in the development of steam boilers seem to be as follows: increase of the unit power, increase of the initial pressure and temperature of steam, application of intermediate steam superheating, mechanized and automated control, manufacture and delivery of boiler equipment in large blocks for easier and quicker assembly.

Nuclear power stations. A unit in which a controlled chain reaction of nuclear fission of heavy elements can be effected is called a nuclear reactor. Fuels for nuclear reactors include natural isotopes (such as U²³⁵) and synthetic isotopes (U²³³, Pu²³⁰, etc.). The nuclear energy liberated by the chain reaction of nuclear fission is transformed into heat which is removed from the reactor by a coolant. A nuclear power station may have one, two or three circuits.

In a single-circuit nuclear power station (Fig. 1.2a), steam is generated directly in the nuclear reactor which therefore doubles as a steam generator. Single-circuit stations are simpler and less expensive and contain the least number of equipment units. On the other hand, the working fluid (water and steam) is irradiated in the reactor and becomes radioactive, which necessitates biological shielding of the reactor proper and the equipment of

the water-steam circuit of the station. Contamination of steam may result in the formation of deposits on the surfaces of the equipment. Since these deposits are radioactive, repairs of the equipment are more difficult to make.

In a two-circuit nuclear power station (Fig. 1.2b), the flow of a liquid, gas or molten metal is heated in the reactor and serves as the heat-transfer agent which then gives up its heat to the working fluid in a steam generator. Therefore, a two-circuit station has an additional unit, a steam generator which naturally increases its cost. The heat-transfer agent must have a certain temperature gradient over the working fluid for the heat to be transferred from the former to the latter. For that reason, with water used as the heat-transfer agent, the temperature of steam entering the turbine is lower than that in a singlecircuit station. The two-circuit scheme requires that the pressure in the reactor be maintained at a higher level than that of the steam delivered into the turbine. On the other hand, twocircuit nuclear power stations have certain advantages over the singlecircuit type in that radioactivity does not extend beyond the first circuit, and therefore, the turbine and other equipment of the second circuit are safely accessible for repairs and biological shielding is indispensable only for the first circuit.

In a three-circuit nuclear power station (Fig. 1.2c), liquid sodium is used as the heat-transfer agent in the first circuit. Being irradiated in the reactor, sodium is liable to activation with the formation of an isotope possessing a high energy of y-radiation. For that reason the first circuit is isolated from the working circuit by an intermediate second circuit. In the second circuit, either sodium or an Na-K alloy can be used as the heattransfer agent. Should the shielding of the second circuit become untight, the radioactive sodium of the first circuit is prevented from getting into the second by maintaining the pressure in the second circuit at a higher level than in the first. In the third circuit, water is the working fluid. In threecircuit nuclear power stations, biological shielding is set up around the first and the second circuit.

Combined steam-and-gas plants and MHD plants. With introduction of supercritical steam parameters ($p = 25.5 \text{ MPa}, t_{rs} = 545 ^{\circ}\text{C}$), secondary intermediate steam superheating $(t_{sec} = 545^{\circ}\text{C})$, heat regeneration, and application of boiler-turbine monobloc units of high power (1 200 MW and more), the thermal efficiency of thermal power stations has approached closely its thermodynamic limit (slightly more than 40%). Any further increase of the initial steam parameters increases substantially the cost of boiler-turbine units, since more expensive high-alloyed steels must be employed. Further, it is difficult to ensure the required reliability of such units.

In view of this, combined systems have been designed and tested industrially. They include a steam-turbine plant and a high-temperature gasturbine plant. Of practical interest are the combined steam-gas plants in which a gas turbine operates in the high-temperature portion and a steam turbine, in the low-temperature portion. Two main possible schemes of combined gas-steam power plants are shown in Fig. 1.3. In both, the gas turbine operates on high-temperature heat. In Fig. 1.3a, this heat is liberated in the combustion chamber into which fuel and compressed atmospheric air are supplied. The gases of combustion are used to perform work in the gas turbine. The exhaust gases and some of fuel are fed into the furnace of a steam boiler which generates steam to drive a steam turbine. The combustion products as fed into the boiler furnace contain around 16% oxygen, so that special air supply for combustion in the boiler furnace is not needed, which makes it possible to dispense with an air preheater. The

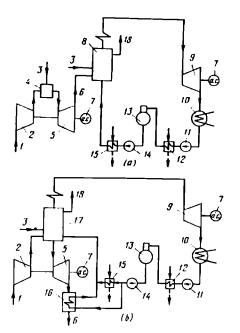


Fig. 1.3. Thermal diagram of a steam-gas power plant

1—air; 2—compressor; 3—fuel; 4—combustion chamber; 3—gas turbine; 6—exhaust gases; 7—electric generator; 8—steam boller; 9—steam turbine; 10—condenser; 11—condensate pump; 12—low-pressure beater; 13—deaerator; 14—feed pump; 15—bigh-pressure heater; 16—heat exchanger; 17—thigh-pressure steam boller; 18—emergency waste gas disposal

unit fuel consumption of steam-gas plants is 3-4% lower than that of a steam-turbine plant with the same initial steam parameters.

Another scheme (Fig. 1.3b) comprises a high-pressure steam boiler in which fuel combustion and heat transfer take place at a high pressure (0.6-0.7 MPa). This makes it possible to intensify these processes and decrease the dimensions of the boiler and thus to save metal substantially. As in the previous scheme, the gas turbine operates on the high-temperature heat of combustion products, i.e. the furnace gases of the high-pressure boiler. The steam generated in the high-pressure boiler is fed into a steam turbine. The combustion products from the gas turbino are cooled by a part of the water flow fed for steam generation. With the same initial parame-

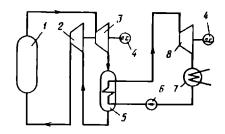


Fig. 1.4. Combined steam-gas power plant on nuclear fuel

1—reactor; 2—compressor; 3—gas turbine; 4—electric generator; 5—steam generator; 6—feed pump; 7—condenser; 8—steam turbine

ters of steam, the unit fuel consumption of a combined steam-gas plant is 4-6% lower than that of a steam-turbine plant. The capital expenditures are also lower by 8-12%.

Combined steam-gas plants with nuclear reactors have also been developed (Fig. 1.4). In this version, the combustion chamber is replaced by a power reactor with a gaseous heat-transfer agent, such as an inert gas, for instance, helium, which allows the temperature at the reactor exit to be raised up to 1500°C or even more. High-temperature gas-cooled reactors can be employed efficiently at nuclear power stations with steam turbines. In steam-gas power plants operating on nuclear fucls, the steam boiler utilizes the heat of exhaust gases of gas turbines.

Another type of combined systems with steam cycle is a magnetohydrodynamic (MHD) plant. Its characteristic feature is that heat is converted into electricity without the use of machines (Fig. 1.5). Atmospheric air is compressed in a compressor, preheated in the boiler to 1 000-1 200°C and fed together with fuel into the combustion chamber where the combustion products form at a temperature of 2 500°C and are ionized. Intensive gas ionization is effected by adding compounds of potassium, caesium and other alkali metals into the combustion chamber.

Hot ionized gases (high-temperaturo plasma), which possess the proper-

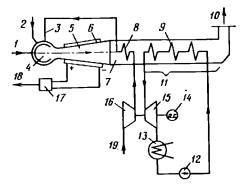


Fig. 1.5. Principal thermal diagram of MHD power plant

1—fuel; 2—ionizing seeds; 3—hot sir; 4—combustion chamber; 5—MHD channel; 6—electric magnets; 7—gas duct; 8—air heater; 9—heating surfaces of steam holier; 10—exit of combustion products; 11—steam holier; 12—pump; 13—condenser; 14—electric generator; 15—steam turbine; 16—compressor; 17—d.c.-a.c. converter; 18—energy to line; 19—air

ties of an electric conductor, are fed through a nozzle into a channel and move in it at a speed of roughly 700 m/s. Powerful permanent magnets create a magnetic field in the channel. As plasma moves in the powerful magnetic field, ionized gas particles induce a direct current in an electric circuit which is then converted into an alternating current. The gas flow leaves the channel at a temperature of 1500-2000°C. This hightemperature heat of the gases is utilized for preheating of the air to be supplied to the combustion chamber and for generation of steam which is fed into a steam turbine. The efficiency of MHD plants may be as high as 50-60%. Roughly 70-80% of the total electric energy are produced in the MHD channel and the remainder, in the steam power plant.

As may be seen from the above principal schemes of electric energy production at power stations, the steam boiler at a thermal power plant and the steam generator at a nuclear power station are indispensable units and belong to the basic units of a power plant of practically any power rating. A steam boiler and steam generator are intended for production of

steam in the required quantity which can ensure the specified power of the turbine and the specified steam parameters.

1.2. Classification of Steam Boilers

According to the laws of phase transformations, the production of superheated steam involves the following sequence of processes: preheating of foed water to the saturation temperature, steam generation, and superheating of saturated steam to the specified temperature. These processes can occur only within strictly defined limits and can be effected in three types of heating surfaces. Water preheating to the saturation temperature is done iu an economizer, the formation of steam takes place in evaporating heating surfaces, and steam superheating is carried out in a superheater.

The working fluid in heating surfaces (water in the economizer, steamwater mixture in evaporating tubes, and superheated steam in the superheater) must move continuously in order to ensure continuous heat removal and maintain the appropriate temperature conditions for the metal of the heating surfaces. In this process, water in the economizer and steam in the superheater come only once in contact with the heating surface (Fig. 1.6). The economizer offers hydraulic resistance to the motion of water, which must be overcome by provision of a sufficiently high head in the feed pump. The pressure developed by the feed pump must exceed the pressure at the entry to the zone of steam generation by the magnitude of the hydraulic resistance of the economizer. Similarly, the motion of steam in the superheater is due to a pressure gradient between the zone of steam generation and the steam turbine.

The combined motion of water and steam in evaporating tubes, which has to overcome the hydraulic resistance of these tubes, can be effected in various ways. Accordingly, a distinction is made between natural-

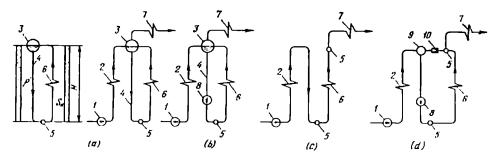


Fig. 1.6. Principal schemes of steam generation in boilers

(a) natural circulation; (b) multiple forced circulation; (c) once-through scheme; (d) combined circulation; 1—feed pump; 2—economizer; 3—drum; 4—downtake tubes; 5—header; 6—evaporating tubes; 7—superheater; 8—pump for multiple forced circulation; 9—mixer; 10—back-pressure valve

circulation boilers, forced-circulation boilers and once-through boilers.

Natural-circulation boilers. Let us consider a closed circuit (Fig. 1.6a) comprising two systems of tubes: a system of heated tubes θ and a system of unheated tubes 4, which all are connected to a drum 3 at the top and to a header 5 at the bottom. This closed hydraulic system of heated and unheated tubes forms a circulation circuit. The volume of the drum occupied by water is called the water space and that occupied by steam is the steam space. The surface that separates the water space from the steam space is called the steam-relieving, or disengagement, area. The water space of the drum and the evaporating tubes are filled with boiler water.

Boiler water is made to boil in the evaporating tubes 6, which thus turn out to be filled with a steam-water mixture of a density ρ_a . Unheated tubes 4 are filled with water of a density p at the existing pressure in the drum. Therefore, the lowermost point of the circuit, i.e. the header, is subjected on the one hand to the action of a water column in the unheated tubes, which is equal to $H\rho'g$, and on the other hand, to the pressure of the steam-water mixture which fills the heated tubes, $H\rho_{a}g$. As steam is formed, the resulting pressure difference $H(\rho' - \rho_a)$ g causes motion of water in the circuit and for that reason is called the driving head in natural circulation:

$$S_{dr} = H \left(\rho' - \rho_a \right) g \tag{1.1}$$

where S_{dr} is the driving head in natural circulation, Pa, H is the height of the circuit, m, ρ' and ρ_a are the densities of water and steam-water mixture, respectively, kg/m³, and g is the acceleration due to gravity, m/s². Since the steam-water mixture in heated tubes moves upward, these are called ascending, or uptake, tubes, while the unheated tubes in which water moves downward, are called descending, or downtake, tubes.

Steam boilers in which the motion of the working fluid in the evaporating tubes is due to the head formed by circulation only which appears naturally on heating the tubes, are called natural-circulation boilers.

In contrast to the motion of water in the economizer and of steam in the superheater, where the working fluid passes through the heating surface only once, the working fluid moves multiply through a circulation circuit. This means that after a single passage through the evaporating tubes water is converted to steam only partially and enters the drum in the form of steam-water mixture. With natural circulation, the mass steam content at the exit from evaporating tubes is 3-25%. With the steam content at the exit, say, 20%, water should be passed five times through the circulation circuit to be vaporized completely.

Since the process of steam formation occurs continuously and feed water is fed to the drum continuously too, as the steam is consumed, water circulates in the circuit all the time and its quantity is not changed. The ratio of the mass flow rate of circulating water G_{ω} , kg/s, to the quantity of steam formed per unit time, G_{ω} , kg/s, is called the circulation ratio (circulation rate):

$$k = G_{w}/G_{\bullet} \tag{1.2}$$

In natural-circulation boilers, the circulation ratio is within 4-30 or sometimes more.

The working fluid can be moved forcedly in evaporating tubes, for instance, by means of a pump included into the circuit. Such plants are called forced multiple circulation boilers (Fig. 1.6b). The driving head of circulation in that case is several times the driving head in natural circulation. This makes it possible to arrange the evaporating tubes in any manner dictated by the boiler design and to effect circulation with the steam-water mixture moving not only vertically upwards, but also horizontally or even downwards. In steam boilers of this type the circulation ratio may be 3 to 10.

Steam boilers with natural multiple circulation feature a drum which is a reservoir providing working fluid circulation in a closed hydraulic system and separation of water from steam. The drum determines all the principal zones of a boiler: economizing, stoam-generating and superheating.

Drum-type boilers operate at subcritical steam pressures, $p < p_{cr}$.

Once-through steam boilers have no drum and the working fluid passes through the evaporating tubes only once (Fig. 1.6c), since the circulation ratio is equal to unity (k = 1). A once-through boiler represents an open hydraulic system. Its another typical feature is that there are no distinct boundaries between the economizing,

evaporating and superheating zones. In the evaporating surfaces of once-through boilers, water is continuously converted to steam. Once-through boilers can operate at subcritical and supercritical pressures, $p \geq p_{cr}$.

In combined-circulation boilers (Fig. 1.6d), back-pressure valve 10 is opened at starting and the plant operates by the circuit as in Fig. 1.6b. On attaining the specified load, the circulation pump is switched off and the back-pressure valve is closed automatically to run the boiler by the scheme of Fig. 1.6c.

1.3. Flow Diagram of Steam Production

The flow diagram of steam production in a steam-turbine power station with once-through boilers fired with pulverized solid fuel is shown in Fig. 1.7. Solid lumpy fuel is delivered in railway cars to the fuel depot. Cars are pushed into dumpers to be turned on the axis through 180° for discharging the fuel into bunkers beneath. Automatic fuel feeders supply fuel onto belt conveyers of the first lift which transfer it to crushers. Crushed fuel (of a size not over 25 mm) is delivered by the second-lift conveyer into the boiler room bunkers. The crushed fuel is then fed into grinding mills where it is ground to the final size and dried. The fuel-air mixture formed in the mills is supplied into the boiler furnace.

In Soviet power engineering, steam boilers of ∏-shaped profile are most popular (for more detail see Sec. 21.1). The boiler consists of two vertical prismatic shafts connected at the top by a horizontal gas duct. The first (larger) shaft serves as the boiler furnace. Its volume may be within a wide range from 1 000 m³ to 30 000 m³ or more depending on boiler capacity and type of fuel. Tubular plane systems, or water walls, are arranged around the entire perimeter and along the whole height of the furnace chamber. They are heated directly by radiant heat of the flame and are essentially

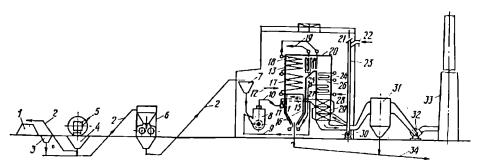


Fig. 1.7. Flow diagram of steam generation

1—coal pile; 2—belt conveyer; 3 and 4—bunkers; 5—car dumper with a car; 6—crushing plant; 7—crusher bunker; 8—coal-grinding mill; 9—primary air; 10—fuel-air mixture; 11—burners; 12—bolter front; 13—succan bolter; 14—furnace space; 15—secondary air; 16—lower radiation section; 17—middle radiation section; 18—upper radiation section; 19—superheated steam; 20—convective superheater; 21—air intake from bolter room; 22—air intake from the outside; 23—cold air duct; 24—reheat superheater; 25—horizontal gas duct; 26—convective shoft (vertical gas duct); 27—convenier; 28—fir horizontal gas duct; 26—forced-draft fan; 31—ash collector; 32—induced-draft fan; 35—stack; 34—slag-ash channel

the radiant heating surfaces. In modern plants, water walls in the furnace are often made of finned tubes which are welded together to form a continuous gas-tight (gas-impermeable) shell, which is covered on the outside by a shell of a heat-insulating material to minimize heat losses to the surroundings; this ensures proper sanitary conditions in the boiler room, and prevents burns of the personnel.

The second vertical shaft and the horizontal duct that connects it with the furnace serve for accommodating heating surfaces which receive heat by convection (convective surfaces) and are called respectively the convective shaft and the convective duct.

Having given up their heat to the water walls, combustion products leave the furnace at a temperature of 900-1 200°C (depending on the type of fuel) and enter the horizontal duct.

As water moves through the boiler tubes, it gradually transforms into steam. The heating surfaces in which steam is formed are called evaporating, or steam-generating. In once-through boilers, the evaporating heating surface is arranged in the lower portion of the furnace and is called the lower radiation section. With supercritical steam parameters, it also includes a radiant economizer. Water supplied to a boiler is called feed water.

Feed water contains certain impurities. During the process of steam generation, the content of steam in the steam-water mixture increases, water evaporates, and the concentration of impurities increases. At a certain concentration at the end of the steamgenerating zone, impurities may be deposited on the internal surfaces of tubes as scale. The conductivity of deposits is only a small fraction of that of the tube metal. This impairs heat transfer to the working fluid and, with intensive heating in the boiler furnace, can cause overheating of the metal which then loses strength and can fracture under the pressure of the working fluid.

The heating surface in which steam generation is completed and steam superheating begins is called the transition zone. Deposition of scale takes place mostly in this zone. In earlier designs of once-through boilers, the conditions of operation of the metal of this zone were made easier by bringing the transition zone out of the furnace into the convective duct where the intensity of heating was roughly one order of magnitude lower (offset transition zone). In modern practice, once-through boilers are fed with practically pure water, so that no scale forms under normal operating conditions and the transition zone can be arranged within the furnace; the working fluid passes from the lower radiation section directly into the water walls above it where steam is superheated (radiant superheater). The radiant superheater can include either two heating surfaces: the medium radiation section and the upper radiation section, which are connected in series, or only the upper radiation section immediately downstream of the lower radiation section. Partially superheated steam flows into the last heating surface which is arranged in the convective duct; this is the convective superheater where steam is heated to the specified temperature. Superheated steam of the required parameters (temperature and pressure) is directed into the turbine. Like any heating surface, the convective superheater is a system of a large number of steel tube coils connected in parallel and interconnected by headers at the inlet and outlet ends.

The temperature of combustion products downstream of the convective superheater is quite high (800-900°C). Part of the worked-off steam can be returned from the turbine for secondary (intermediate) superheating, usually to the same temperature as that of steam from the main superheater. This is the intermediate (reheat) superheater (or, simply, reheater).

The combustion products at the outlet from the intermediate superheater are still rather hot (500-600°C) and their heat can be utilized in a convective economizer. Feed water supplied into the convective economizer is preheated to a temperature below the saturation point and is fed into the lower radiation section. The temperature of combustion products downstream of the economizer is 300-450°C or sometimes more. Further heat utilization is effected in a next convective heating surface, the air heater. It is a system of vertical tubes, with combustion products flowing in the tubes and air, between them. The temperature of air is 30-60°C at the inlet to the air heater (cold air) and

250-420°C at the outlet (hot air), depending on kind of fuel and method of combustion.

With pulverized fuel combustion, the preheated air is separated into two flows. The primary air is used for drying of fuel and transport of fuel dust through burners into the boiler furnace. The temperature of this fuel-air mixture is 70-130°C. The secondary air is directed immediately through burners into the furnace (bypassing the fuel mills) at the temperature it has had after the air heater.

Downstream of the air heater, the combustion products have already a rather low temperature (110-160°C). Further utilization of their heat is economically inefficient and they are ejected through the stack into the atmosphere. They are called waste, or chimney, gases.

Upon burning, fuel leaves fly ash which is mostly carried off by combustion gases. Fly ash is catched (collected) in a fly-ash collector which is arranged upstream of the induced-draft fan. This arrangement prevents abrasion wear of the induced-draft fans and contamination of the atmosphere with fly ash. The collected ash is removed by means of ash-removal devices. Part of ash falls onto the bottom of the boiler furnace and is removed continuously by the ash-handling system.

The flow diagram of steam generation in drum-type boilers differs from that described above only in the design and operation of the boiler proper (Fig. 1.8). In this case, the steamwater mixture formed in the furnace water walls is fed into the boiler drum. The steam separated in the drum is practically dry and is fed first into the superheater and then into the turbine.

As follows from the flow diagram of steam generation (see Fig. 1.7), a boiler plant has the following basic paths:

—the fuel path, i.e. a combination of elements in which solid fuel is transported, crushed, ground, and delivered to the boiler furnace for combustion.

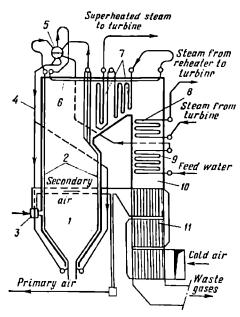


Fig. 1.8. Diagram of free-circulation drumtype boiler

I—furnace apace; I—water walls; I—burners; downtake tubes; 5—drum; 6—radiant superheater; 7—convective superheater; 8—intermediate superheater (reheater); 9—economizer; 10—convective gas duct; 11—air heater

The fuel path comprises the crushing equipment, conveyers, crushed-fuel bunker, grinding mill, and pulverized-fuel duct leading to the furnace. Up to the crushed-fuel bunkers, fuel is transferred by conveyers. Beginning from the grinding mill, the resistance in the fuel path is overcome by the head of the fan;

—the water-steam path, or circuit, is a system of series-connected elements for the transport of water, steamwater mixture and superheated steam. The water-steam circuit includes the

following elements: an economizer, furnace water walls, and steam superheaters;

—the air path includes a combination of elements for suction of atmospheric (cold) air, its preheating, transport and supply into the furnace. The air path comprises a cold air duct, air heater (its air side), hot-air duct, and burners;

—the gas path is a complex of elements in which the combustion products flow from the furnace into the atmosphere; it begins in the boiler furnace and passes through the superheaters, economizer, air heater (its gas side), ash collector and stack.

The air and gas paths are connected in series forming what is called the gas-air path. The transition from one to the other takes place in the boiler furnace space. The diagram of a gasair path is shown in Fig. 1.9a. In this circuit, air is transported by blowers and the corresponding air path in the portion between the blower and furnace is at a pressure higher than the atmospheric. Combustion products are transported by induced-draft fans arranged downstream of the boiler, and therefore, the furnace proper and all gas ducts are at a pressure below the atmospheric. This is what is called the balanced draft scheme.

The transport of air to the furnace and of combustion products to the atmosphere can be ensured by forced-draft fans only, i.e. without induced-draft fans (Fig. 1.9b). In that case, the furnace and gas ducts are under a certain excess pressure (supercharged). For comparison, Fig. 1.10 shows the pressure distribution in the gas-air path

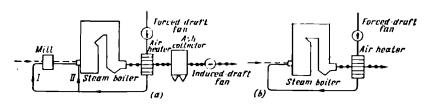


Fig. 1.9. Diagrams of gas-air paths

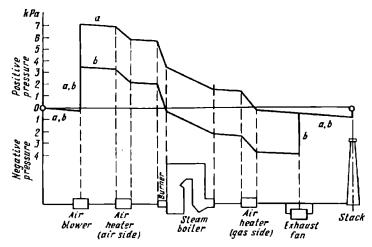


Fig. 1.10. Pressure distribution in gas-air path of (a) supercharged boiler plant and (b) balanced-draft plant

of a boiler with balanced draft and in that with a supercharged furnace.

1.4. Principal Characteristics of Steam Boilers

The steam-generating capacity D, t/h (or kg/s), is the quantity of steam produced by a boiler per unit time. A boiler is calculated for a rated steam-generating capacity D_r , which is understood as the highest load of the boiler in stable operation for a long time on the specified fuel and with the rated parameters of steam and feed water. The steam-generating capacity of industrially made boilers ranges within very wide limits.

At present, power engineering in the USSR is based on the use of boiler plants producing 1 000, 1 650 or 2 650 t/h of steam of supercritical parameters (pressure 25.5 MPa, steam superheating 545°C, intermediate superheating 545°C) and having the efficiency of 92-94%. Such boilers supply steam to turbines of a power respectively of 300, 500 or 800 MW. The steam boiler and the turbine constitute a monobloc unit. Recently a monobloc unit for a power of 1 200 MW with a boiler of a capacity of 3 950 t/h has been put into operation. At some

heat and power stations, smaller boilers for lower steam parameters are employed.

Superheated steam is characterized by its parameters: the pressure and temperature in the outlet header of the superheater. The boilers employed at power stations differ in the pressure of steam, which may be high (10 or 14 MPa) or supercritical (25.5 MPa). Steam boilers for pressures of 14 MPa and more are as a rule made with secondary steam superheating.

Boiler preassembling. The time of construction and mounting of boiler plants can be shortened appreciably by making larger (preassembled) boiler units at the manufacturing works. Preassembled units must match with the dimensions of railway cars. Preassembled units are transported to the boiler plant site to be mounted into a boiler. The coefficient of preassembling, which is the ratio of the total mass of preassebled units to the whole mass of a boiler plant, may be as high as 80-90%. The largest difficulties arise in manufacture of preassembled units of the boiler structure. Preassembling places certain requirements on the boiler design, since preassombled units should satisfy the special conditions of their transport and mounting.



POWER-PRODUCING FUELS AND THEIR CHARACTERISTICS

2.1. Kinds and Compositions of Fuels

The development of power engineering is directly linked to the construction of new thermal and nuclear power stations, i.e. plants operating on organic or nuclear fuels.

Organic fuels are those which can generate substantial quantities of heat (per unit mass or unit volume) by reacting with oxygen.

Organic fuels suitable for production of large quantities of heat with a sufficient economy are termed powerproducing. Their reserves must be enormous and relatively easily explorable. Besides, they must be of low value as starting materials for other branches of industry. The most popular kinds of power-producing fuels employed at thermal power stations are as follows: solid—coals and lignites and some products prepared from them, anthracite and semianthracite, liquid—fuel oil, and gaseous natural gas. Peat, oil shales, stabilized oil and industrial gaseous fuels (blast-furnace and coke-oven gas) are used to a lesser extent, though in some regions of the country they constitute an appreciable share of the fuel balance.

Of late, electric energy is produced more and more at nuclear power stations which utilize the energy of nuclear fission of radioactive heavy metals: uranium (U²³⁵) and plutonium (Pu²³⁹). The richest uranium ore, uraninite, contains 65-90% of uranium dioxide UO₂ in which only 0.72% falls on radioactive U²³⁵ and the balance is the common U²³⁸. The original nuclear fuel is processed in gas-

diffusion plants to raise the concentration of U²³⁵ to 1.5-3.5%, after which it can be charged into nuclear reactors. One kilogram of U²³⁵ liberates on fission roughly 85 mln MJ of heat, which is equivalent to the combustion of 3 500 t of coal with the heating value 24.5 MJ/kg.

In the USSR, thermal power stations consume roughly 40% of the total organic fuel. Coals, fuel oil and natural gas are the predominant kinds of organic fuel in their fuel balance. The share of coals burned at thermal power stations increases gradually due to the exploitation of new coal fields in Siberia and Northern Kazakhstan. The consumption of fuel oil and natural gas is approximately at the same level as coal. Other kinds of solid fuel, such as peat and oil shales, constitute only 6-7% of the total fuel consumption by thermal power stations. In new coal fields, less expensive opencast mining will be employed increasingly.

All fossil fuels, i.e. solid fuels and petroleum, have formed in the process of long transformation of the original vegetable mass and died-off animals under a layer of earth or water. The process occurred at a different rate as regards the carburization of the fuel, i.e. increasing carbon concentration and decreasing concentration of oxygen and hydrogen (Fig. 2.1).

The degree of carburization, which characterizes the depth of chemical transformations in fuel (which is called the chemical age of fuel) is not determined directly by its geological age, i.e. the duration of carburization process.

Fuel	c.º %	o°. %	H ^c . %	v ^c , %
Wood	50	62.5	6	85
Peat	51	39	6	55
Brown coal	75		5	37
Coal	90	\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\		g
Anthracite				且
	93	2	2	4

Fig. 2.1. Comparison of elemental analyses of principal fuels

Crude petroleum is a mixture of organic compounds, including minor quantities of liquid sulphur and nitrogen compounds, paraffins and resins. Upon distillation of light fractions and oils (petrol, naphta, kerosene, gas oil, straw oil), there remain viscous heavy fractions—fuel oil which is used as a liquid power-producing fuel. The mineral impurities present in the crude petroleum are mainly concentrated in fuel oil.

Natural gases either form together with petroleum or are produced by synthesis in the presence of water and metal carbides at large depths under the action of high pressure and temperature. In many cases, natural gas accompanies petroleum. This is what is called a casing-head gas; it can be used as a power-producing fuel.

Replacement of solid fuels by liquid and gaseous improves the operating conditions of power stations and can decrease sensibly the cost of the equipment and increase its efficiency. For instance, the capital expenditures for the construction of a power station to be fired with gas or fuel oil are lower by 20-24% than those for a solid-fuel fired station of the same power. The efficiency of fuel oil-fired plants is 4% higher (in terms of electric energy produced) than that of a solid-fuel fired plant.

The explored reserves of natural gases and petroleum are however limited and constitute only 6% of the total world reserves of organic fuel. Besides, natural gases and petroleum are of the highest value as starting

materials for various branches of national economy. On the other hand, the reserves of coal exceed 71% of the total explored reserves of fuel in the world and therefore coals are the principal organic fuel.

The organic mass of solid and liquid fuels consists of a large number of complex compounds of the principal five elements: carbon C, hydrogen H, oxygen O, sulphur S, and nitrogen N. Besides, any fuel contains mineral impurities A, which enter the original bed mainly from the outside, and moisture W. For that reason, the chemical analysis of solid and liquid fuels is determined not in the terms of their compounds, but as the total mass of chemical elements in 1 kg of fuel, i.e. the elemental analysis of a fuel is determined.

One should distinguish between the following five elemental masses of fuels:

working mass

$$C^{\omega} + H^{\omega} + O^{\omega} + N^{\omega} + S^{\omega}_{p} + A^{\omega} + W^{\omega} = 100 \%$$
(2.1)

analytical mass

$$C^a + II^a + O^a + N^a + S^a_b + A^a + W^a = 100 \%$$
(2.2)

dry (moisture-free) mass

$$C^d + \Pi^d + O^d + N^d + S_v^d + A^d = 100 \%$$
(2.3)

resolved combustible mass

$$C^c + 11^c + 0^c + N^c + S_0^c = 100 \%$$
 (2.4)

and organic mass

$$C^{o} + 11^{o} + 0^{o} + N^{o} + S^{o} = 100 \%$$
 (2.5)

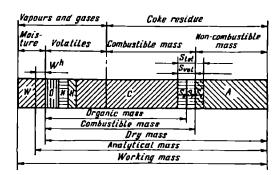


Fig. 2.2. Diagram of elemental analysis of solid fuel

The working mass of fuel is the mass in the form it is delivered to the plant. The consumption of fuel and the volumes of combustion products formed are calculated in terms of the working mass. The working fuel upon grinding to pulverized state and when dried in laboratory to the air-dry state, loses the free moisture, and its mass is then called analytical. The remaining moisture in fuel, W^a , which is combined in the original substance, is more often called hygroscopic moisture, i.e. $W^a = W^b$.

Upon heating of fuel to 102-105°C, all its moisture evaporates, thus leaving the dry mass of fuel. The combustible mass of fuel includes the elements of the original organic matter and the sulphur of inorganic combustible compounds (for instance, pyrite FeS₂); for that reason, it is called the resolved combustible mass.

In equations (2.1) to (2.4), S_v is the volatile sulphur, i.e. the sum of pyrite sulphur and organic sulphur which can be oxidized in the boiler furnace: $S_v = S_p + S^o$.

The organic mass of fuel is essentially its combustible mass plus pyrite sulphur. Apart from the two kinds of sulphur mentioned, there is also the sulphate sulphur S₄ which enters the composition of higher oxides (such as CaSO₄) and cannot be further oxidized in combustion. The various masses of solid fuel are shown diagrammatically in Fig. 2.2. It should be distinguished between the external and internal ballast in the composition

of a fuel. The former includes moisture and mineral matter and the latter includes oxygen and nitrogen which enter the original organic matter.

Combustible elements in fuel are carbon, hydrogen and sulphur, with carbon being the principal combustible element. It has a high heating value (34.1 MJ/kg) and constitutes the major portion of the working mass of fuel (50-75% in solid fuels and 83-85% in fuel oils). Hydrogen has a very high heating value (120.5 MJ/kg), but its content in solid fuels is not high (H = = 2-4%) and is somewhat higher in liquid fuels (10-11%). Sulphur has a low heating value (9.3 MJ/kg) and is present in fuels in minor quantities $(S^w = 0.3-4\%)$ and for that reason is of low value as a combustible element. The presence of sulphur oxides in combustion products increases the risk of corrosion of metallic heating surfaces and in certain concentrations may be dangerous to vegetation and animals; this necessitates measures for the collection of sulphur from waste gases. According to their sulphur content, fuel oils are low-sulphur divided into < 0.5%), medium-sulphur (S^w = 0.5-2%) and high-sulphur (S $^{\omega} > 2\%$) grades.

In contrast to solid and liquid fuels, gaseous fuels are mechanical mixtures of combustible and non-combustible gases. Natural gases consist preferably of methane CH₆ (up to 90-96%), with minor quantities of heavier hydrocarbons (ethane C₂II₆, propane C₃H₆,

butane C_4H_{10} , etc.) which are often written generally as C_mH_n (1-6%). Besides, natural gas contains minor quantities of non-combustibles: nitrogen N_z (1-4%) and carbon dioxide CO_z (0.1-0.2%).

2.2. The Heating Value and Resolved Characteristics of Fuels

The quantity of heat liberated on combustion of a unit mass or volume of fuel is the principal thermal characteristic of fuels, called the heating, or calorific, value. One should distinguish between the upper and lower heating value. The upper heating value Q_u is the quantity of heat liberated on combustion of 1 kg of solid or liquid or 1 m³ of gaseous fuel under the conditions that water vapours are condensed and the combustion products are cooled to 0°C. The lower heating value Q_1 differs from the upper heating value by the heat of evaporation of the moisture present in the fuel and the moisture formed on combustion of hydrogen. In power plants, moisture of the combustion products remains in a vaporized state and the heat consumed for its evaporation is lost. With a higher moisture content of fuel, Q_l is lower.

The lower heating value, kJ/kg, can be found by the formula:

$$Q_1 = Q_u - Q_w \tag{2.6}$$

In the general case, the heat of moisture condensation, kJ/kg, is:

$$Q_{W} = 2500 \left(\frac{9H}{100} + \frac{W}{100} \right) = 225H + 25W$$
(2.7)

where H and W are the concentrations of hydrogen and moisture, %, and $2\,500$ is the heat of condensation of 1 kg of moisture at atmospheric pressure, kJ/kg.

When finding the lower heating value for other masses of fuel, except for the analytical and working mass, formula (2.7) becomes simpler because the moisture term is excluded:

$$Q_{w} = 225 \text{ H}$$

The higher heating value of solid and liquid fuels can be determined experimentally by burning a sample of fuel in a calorimetric apparatus.

The heating value of fuels can be determined approximately from their elemental analysis.

In this respect, the most simple and accurate are Mendeleev's formulae which contain empirically found coefficients for various combustible elements. For instance, the formula for determining the lower heating value of the working mass of solid and liquid fuels is as follows:

$$Q_i^w = 339C^w + 1 \ 030H^w - 109 \ (O^w - S^w) - 25W^w$$
 (2.8)

where C^w, H^w, etc. are the concentrations of elements in the working mass of fuel, %. For gaseous fuels, if their analysis is known exactly, the heating value of 1 m³ of dry gas can be determined by the formula:

$$Q_l^d = 0.01 \ (Q_{\rm H_2} II_2 + Q_{\rm CO}CO \\ + Q_{\rm CH_4} CII_4 + Q_{\rm C_2H_6} C_2 II_6 + \ldots) \ \ (2.9)$$
 where H₂, CO, CH₄, C₂II₆, etc. are the volume concentrations of combustible gases in fuel, %, and $Q_{\rm H_4}$, $Q_{\rm CO}$, $Q_{\rm CH_4}$, $Q_{\rm C_1H_4}$, etc. are their heating values, kJ/m³.

Steam boilers of the same steamgenerating capacity can consume widely different quantities of fuel, since the heating value of various fuels may vary within wide limits. For comparison of the efficiency of various power plants and for simpler calculations of the combustion of various fuels, the concept of reference fuel has been introduced. It is a conditional fuel whose heating value is $Q_r =$ = 29.33 MJ/kg (7 000 kcal/kg). The consumption of various kinds of fuel at power plants can be calculated in terms of the reference fuel by the formula:

$$B_r = B \frac{Q_l^w}{Q_r} \tag{2.10}$$

where B_r and B are the consumption of the reference fuel and natural fuel, respectively.

The method of expressing the clemental analysis of fuels and their external ballast (moisture and ash content) as percentage of the original mass of burned fuel is used widely in power engineering. This method, however, is not always convenient for the analysis of operating conditions of steam boilers. For instance, as the external ballast of a fuel increases, the heating value of the fuel becomes lower, and therefore, more fuel must be burned to maintain the same steam-generating capacity of the boiler. As a result, the mass of the ballast entering the furnace increases much more appreciably than the fuel consumption, which can form intolerable conditions for boiler operation. Thus, the percentage of moisture, ash or sulphur in fuel is not sufficient to characterize the valuability of a fuel.

The mass consumption of fuels burned in steam boilers can be characterized more reliably by relating the concentration of chemical elements and ballast, per cent, to the unit of the lower heating value, 1 MJ, which is called the resolved characteristic of fuel.

The resolved moisture content, ash content and sulphur content (% kg//MJ) are determined by the formulae:

$$W^{r} = \frac{W^{w}}{Q_{l}^{w}}, \quad A^{r} = \frac{A^{w}}{Q_{l}^{w}}, \quad S^{r} = \frac{S^{w}}{Q_{l}^{w}}$$
(2.11)

For instance, with the same original sulphur content $(S^w = 3\%)$ of fuel oil $(Q_i^w = 39 \text{ MJ/kg})$ and brown coal $(Q_i^w = 12 \text{ MJ/kg})$, the mass of sulphur oxides carried off with combustion products is in the latter case 3.25 times higher, according to the ratio of the resolved sulphur contents: $S^r = 0.077$ of fuel oil and $S^r = 0.25$ of brown coal.

2.3. Technical Characteristics of Solid Fuels

Efficient combustion of fuels in steam boilers requires knowledge and correct account of a number of deci-

sive characteristics of fuels which, apart from the heating value, include the ash content, moisture content, the yield of volatiles, etc.

Ash content. Mineral impurities are present in all kinds of solid fuel. Their major portion is not associated with the organic mass of fuel. As regards their origin, mineral impurities can be divided into internal which accumulate in fuel beds in the course of fuel formation and external which pass into the fuel from the surrounding gangue during exploitation.

As fuel is burned, its mineral components are subjected to high-tomperature transformations. Complex mineral compounds of the type of clays $Al_2O_3 \cdot 2SiO_2 \cdot 2H_2O$, feldspars K_2O . ·Al₂O₃·6SiO₂, sulphates and carbonates CaSO₄·2H₂O·CaMg(CO₃)₂, etc. are destroyed and partly afteroxidized by atmospheric oxygen. The residue remaining after the combustion of fuel, i.e. ash, consists mainly of a number of oxides: SiO₂, Al₂O₃, Fe₂O₃, CaO, MgO, K2O, Na2O and its mass turns out, on the average, to be 10% smaller than the original mineral mass of fuel. The percentage of the ash residue relative to unit mass of original fuel is called the ash content.

The properties of ash play an important part in the operation of a steam boiler. Finest solid particles of ash are entrained by the flow of furnace gases and carried off from the furnace; this is what is called fly ash. Another part of ash is melted in the flame core and drops to the furnace bottom or sticks to the surrounding furnace walls and forms slags on solidification, i.e. solid solutions of minerals whose composition may differ from that of fly ash.

Of special importance for the fuel combustion process are the fusibility characteristics of ash. The melting points of various minerals and their alloys differ widely, ranging from 600° to 2 900°C. For that reason, ash is not melted at a fixed temperature, but is softened gradually and changes from the solid to liquid state with

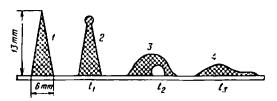


Fig. 2.3. Cone test of ash
1-before heating; 2-beginning of deformation; 3-softening; 4-fluid state

increasing temperature. The melting temperature of ash is determined by the standard cone test, i.e. a cone of standard size is pressed from ash and placed into a furnace (Fig. 2.3). During heating, the following characteristic temperatures are marked:

 t_1 is the beginning of deformation, when the shape of the cone starts changing, $t_1 = 1000-1200$ °C;

 t_2 is softening, when the top of the cone drops to the base or assumes a droplet-like shape, $t_2 = 1\ 100\text{-}1\ 400^{\circ}\text{C}$; and

 t_3 is the fluid state, when ash begins to spread over the plane base, $t_3 = 1 200-1 500$ °C.

The basic characteristic of slag is its viscosity.

Molten slag, if it is in an actual liquid state, flows freely along a vertical or inclined wall when its viscosity is less than 200 P (poise). The temperature of molten slag at which it can flow out freely from a hole is called the temperature of normal liquid slag removal, t_{sr} . The temperature of fusion of ash and the typical coefficients of viscosity can be found from tables of power-producing fuels.

In burning, the major portion of mineral composition of fuel is transformed into fine fly ash which is carried off by the gas flow. In furnaces operating under different thermal con--ditions of combustion and slag removal, the amount of fly ash carry-over, a_c , may range from 0.85-0.95 to 0.2-0.4. The remaining portion drops down onto the furnace bottom as slag and is removed therefrom: $a_{st} =$ $= 1 - a_c$. With a higher ash content of fuel and a higher concentration of fly ash in furnace gases, more intricate and expensive ash-collecting devices are needed to prevent pollution of the air basin. In that case, the speeds of gases in convective ducts of boilers are reduced to avoid abrasion wear of tubes, while deposition of ash particles on the heating surfaces impairs the conditions of heat transfer. As a result, the boiler plant becomes too bulky in design.

The yield of volatiles and coke residue. If a sample of dry solid fuel is placed into a crucible and heated gradually in an inert medium without air, its mass will decrease. At high temperatures, oxygen-containing molecules of the fuel dissociate and form gaseous substances which are called volatiles (CO, 112, CII4, CO2, etc.). The evolution of volatiles from solid fuels takes place in the temperature range 110-1 100°C. The highest yield (up to 95%) occurs at a temperature near 800°C (Fig. 2.4). For that reason, the yield of volatiles from solid fuels is determined conditionally as the decrease of the mass of fuel sample upon holding it in a crucible at $t = 850 \pm 25$ °C for 7 minutes, related to the combustible mass of fuel, V° %.

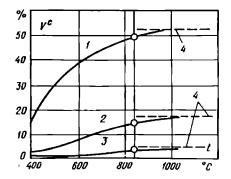


Fig. 2.4. Evolution of volatiles depending on temperature

1—brown coal; 2—lean coal; 3—anthracite; 4 ultimate yield of volatiles for particular fuel Since the yield of volatiles is determined in the first place by the concentration of oxygen in fuel, it is higher in 'younger' fuels (Fig. 2.1). For instance, the yield of volatiles is $V^c = 45-50\%$ from brown coals, 25-40% from coals, and only 3-4% from anthracites.

The solid combustible residue remaining upon evolution of volatiles is called coke. It may be either dense (sintered) or loose (powdered). In air, coke ignites at a temperature of 900-1200°C. Volatiles evolving from the fuel ensure earlier ignition of coke, since they can ignite at a lower temperature than the coke residue (350-600°C) and thus raise quickly the temperature of coke particles. Their effect is especially strong at the initial stage of fuel burning. Fuels with a higher yield of volatiles ignite more quickly and burn more completely.

Moisture content. It is distinguished between the adventitious, adsorbed, cellular, and inherent moisture. All kinds of moisture, except for inherent moisture, are removed from fuel on heating to 102-105°C. The inherent, or hydrate, moisture is firmly bonded with the mineral portion of fuel and enters the composition of crystals of the substance.

Solid fossil fuels contain mostly adsorbed moisture which is determined by the adsorptivity of complex colloids of the organic mass of fuel. The highest adsorptivity is exhibited by peat, brown coals and some youngor coals. The adsorptivity of a fuel determines its hygroscopic moisture content Wh. This moisture characterizes indirectly the ago of fuel: it is lower in older fuels. For instance, $W^h = 10-13\%$ in brown coals and only 1.5-2.5% in anthracites. Knowledge of W^h is essential for estimating the admissible moisture content of pulverized coal to avoid sticking of particles (at an excessive moisture content) or explosion of overdried dust.

The adventitious, or mechanically retained moisture appears in fuel upon

contact with water and remains on its surface due to wetting. Its magnitude depends on the particle size of fuel and the external conditions during transport and storage. Cellular, or capillary, moisture is determined by the porosity of fuel structure. It is most pronounced in peat.

high moisture content in the working mass of fuel may cause serious difficulties in fuel combustion. It decreases the heating value of fuel, increases fuel consumption and the volume of combustion products, and involves higher heat losses with waste gases and greater energy consumption for driving the induceddraft fans. An elevated moisture content of furnace gases can cause stronger corrosion of metal in the air heater and increase contamination of the heating surfaces. Wet fuel is sticky, which involves difficulties in its transport and preparation, and besides, it can congeal in winter time. The effect of the sulphur present in fuel on the boiler operation will be discussed in the section to follow.

Table 2.1. Grading of Coals

Coal type	Grading	Yield of volatiles, V ^c , %	Characteristic of coke residue
Long-flame coal	D	36 and more	Powdered, poorly sinte- red
Fiery coal	G	36 and	-
Fiery fat	GZh	more 31-37	_
Fat coal	Zh	24-37	_
Fat coking	KZh	25-33	Densely sintered
Coking coal	K	17-33	_
Leaned ca- king coal	S	14-27	-
Low-caking coal	SS	17-37	Poorly sin- tered, pow- dered
Lean coal	Т	9-17	Poorly sin- tered, pow- dored

Grading of solid fuels. Solid fuels are graded mainly according to their moisture content in the working mass (brown coals) or the yield of volatiles (coals). For instance, brown coals are divided into three groups: B1 with moisture content \tilde{W}^{w} up to 40%; B2 with $W^{\omega} = 30-40\%$, and B3 with W^{w} less than 30%. The grading of coals is based on the yield of volatiles and the characteristics of coke (Table 2.1).

Fine fractions of fuel (screenings) which remain after screening of the produced fuel are additionally graded by a letter that shows their particle size, for instance: Sh - fractions from 6 mm and less; SSh-fractions from 13 mm and less; R-ordinary (unscreened) fuel, etc.

2.4. Technical Characteristics of Fuel Oil and Natural Gases

The quality of fuel oil has great effect on its combustion in boiler furnaces and the scheme of fuel oil preparation and supply to a power station.

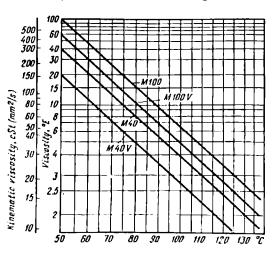
Viscosity. Viscosity is lone of the principal technical characteristics of fuel oils and the basis for their grading. Fuel oils are divided into light, medium and heavy grades. Light grades include marine fuel oils (F5 and

Fig. 2.5. Effect of temperature on the viscosity of fuel oil

F12), while medium and heavy grades are used as furnace fuel in boilers and other stationary power plants and in process plants. Depending on their viscosity and other physical characteristics, furnace fuel oils are subdivided into the following grades: highquality fuel oils 40V and 100V and furnace fuel oils 40 and 100 (40V and 100V are heavy grades of fuel oil) [9].

Viscosity of fuel oils is measured in units of kinematic viscosity (centistokes, cSt) or in degrees of Engler viscosity (°E) as measured in an Engler viscometer by the time of flow of a portion of fuel oil from a calibrated hole at standard temperature (80°C for heavy fuel oils). For normal transportation through pipelines and for fine atomization of fuel oil in burners, its Engler viscosity should be within 2-3.5°E. The viscosity of fuel oil heavily depends on temperature (Fig. 2.5). Its variations with temperature are due to the presence of paraffinic hydrocarbons in fuel oil. For easy transportation in pipes and for normal operation of fuel oil pumps, the temperature of fuel oil should be maintained near 60-70°C.

Rheologic properties. At low temperatures (10-25°C) viscous fuel oils can stick to the surfaces of vessels, pipes, etc. and remain on them in a layer whose thickness is greater with



a lower temperature. This effect is due to the rheologic property of fuel oil, i. e. to the rearrangement in the structure of hydrocarbon molecules which takes place on a decrease of temperature. Sticking of fuel oil is avoided by heating it to a temperature of 70°C or more.

Density. The density of fuel oils is usually measured in relative units, i.e. as the ratio of the fuel oil density to the density of water at 20°C which is $\rho_{20}=0.99\text{-}1.06$. With an increase of temperature, the density of fuel oils decreases and can be found by the formula:

$$\rho_t = \frac{\rho_{20}}{1 - |-\beta|(t - 20)} \qquad (2.12)$$

where ρ_t , ρ_{20} is the relative density of fuel oil at a given temperature and at 20°C and β is the coefficient of volume expansion of fuel oil on heating by 1°C; for fuel oils, $\beta = (5.1-5.3) \times 10^{-4}$.

Ash content. During processing of crude petroleum, the mineral impurities present in it are mainly concentrated in heavy fractions and especially in fuel oil. The ash residue remained upon combustion of fuel oil is not large, not more than 0.1% of the dry mass. Fuel oil ash is characterized by a certain content of vanadium whose concentration may be as high as 50% or even more.

Moisture content. Fuel oils usually contain 1-3% water. Their moisture content can rise substantially (up to 10-15%) in the course of fuel oil preheating before pouring it from tank cars, owing to the condensation of lowpressure steam which is used for heating. A small concentration of moisture in fuel oil is favourable for its atomization in burners and improves inflammability characteristics. With an elevated concentration of moisture, there is a high risk of corrosion of convective heating surfaces; this also increases the loss of heat with combustion products.

Sulphur content. Petroleum and solid fuels contain sulphur in the form of complex sulphur compounds. When petroleum is processed, the major portion of sulphur compounds (70-90%) passes to high-boiling fractions which are the main components of fuel oil. During combustion of fuel oil or solid fuel, sulphur is oxidized to SO₂ and a minor portion of it can form the higher oxide SO₃ (if there is enough oxygen in the combustion zone), which forms a corrosive medium on low-temperature heating surfaces. The content of sulphur in fuel oils is roughly the same as in solid fuels $(S^{\omega} = 0.5-3\%)$, but the corrosive ability of the gaseous medium that forms on the combustion of fuel oil is several times higher. This is due to the fact that, unlike fuel oil, solid fuels contain certain components in the ash which can neutralize acid media.

Congelation temperature. The congelation temperature of petroleum products is the temperature at which they become so thick that remain in place and do not flow out for 1 minute from a test glass inclined at 45°. High-sulphur fuel oils with a high concentration of paraffins (Grades M-100 and M-100V) are characterized by a high congelation temperature (25-35°C). The congelation temperature of fuel oil should be considered properly when selecting the scheme of its transport and storage.

Flash point. The flash point is the temperature at which a mixture of fuel oil vapours and air can be ignited when it comes in contact with an open flame. Fuel oil grades used at power stations have a flash point of 90-140°C, while high-paraffinic fuel oil may have a lower flash point (up to 60°C); the flash point of crude petroleum is only 20-40°C. In order to avoid fire hazard, preheating of fuel oil in open systems should be carried out at temperatures below its flash point and in all cases, not above 95°C to prevent holling of the moisture which may be present in the bulk of fuel oil.

The basic technical characteristics of natural gases are density, explosiveness and toxicity.

Fig. 2.6. Ignition ranges of gas-air mixtures at 20° C (p = 0.1 MPa)

Gas	Førmula	Jn	fla	mmu 6 by	ibil vol	ity ume	limit P	s,	
Hydrogen	Hz	Ø	////		177	222	1111	77	1
Carbon monoside	СН			1		•		22:2	
Methane	C H	Ø						Т	
Ethane	C2H6		5						_
Propane	C ₃ H ₈	Ø	Γ						
Butane	C, H _{f0}	<i>[</i> 2]							_
Ethylene	C2H,	Ø	///	72					
Acetylene	C ₂ H ₂		///	17.17	277	(///	1111	7	_
Hydrogen sulphide	H ₂ S	17	///	<u> </u>	77				
Natural yas		172	5	Π			i		
Blast-Furnace gas					\mathbb{Z}	///	7777	<u> </u>	
Coke-oven gas		Z	////	////				$oxed{\mathbb{I}}$	
	Volume in gas	pro s-ai	pori	ro tion ixtu	of g	0 7as %	60		(

Density. Almost all kinds of gaseous fuel are lighter than air, so that escaped gases may collect under roofs. Before firing a boiler, it is essential to check that there are no gases in places of their probable accumulation. Various gases are compared by using the concept of relative gas density which is the ratio of the density of a given gas under standard conditions (0°C, 1015 Pa) to the density of air:

$$\rho_r = \rho_g/\rho_a = \frac{\rho_g}{1.293} \qquad (2.13)$$

where ρ_g and ρ_a are the densities of gas and air respectively under standard conditions, kg/m³.

Explosiveness. A mixture of a gas and air in a certain proportion can explode when in contact with open flame or even a spark, i.e. it ignites and burns at a speed near the velocity of sound propagation. Explosive concentrations of combustible gases in air depend on the composition and properties of a particular gas (Fig. 2.6). In concentrations below the lower limit of inflammability (explosiveness), a gas-air mixture cannot burn. In concentrations above the upper in-

flammability limit, a gas-air mixture burns without explosion.

Toxicity. Toxicity is the ability of gases to poison the living organisms. In that respect, carbon monoxide CO and hydrogen sulphide H₂S are most dangerous.

Since all components in natural gas are intermixed evenly, the concentration of harmful gases in air can be determined by the presence of methane whose concentration is measured by methanometer. The test determines the explosiveness of the gas mixture. Almost all natural gases are odourless. For easier detection of gas leakages and taking proper safety measures, natural gas is odoured before pumping into a gas pipeline, i.e. a substance having a strong smell is added to it.

2.5. Main Deposits of Fossil Fuels

Fossil solid fuels are distributed over the USSR territory extremely unevenly. The most developed industrially regions in the European part of the country are not rich in fuel. Of greatest importance here is the Donetsk coal basin which possesses various grades of coal and anthracite, but its reserves can no more satisfy the growing demand. Besides, coal seams are thin and deep-lying which makes coal production too expensive.

The main mass of coals is concentrated in the Central and West Siberia and Kazakhstan. These coals are cheaper than Donetsk coal, notwithstanding the costs required to transport them to the European regions. Further, there are large reserves of brown coals in the Kansko-Achinsky basin (Central Siberia), with powerful shallow-lying seams which can be produced profitably by open cast technique; this is the cheapest fuel in the USSR. Similar characteristics have Ekibastuz coals (East Kazakhstan). Kansko-Achinsky brown coals will be processed by a complex technology into valuable chemical products, brown-coal fuel oil and coke breeze, a fuel of a high heating value (around 29.3 MJ/kg).

Petroleum fields in Tyumen district are being exploited intensively. The production of petroleum and condensed gas in this region amounts to about 50% of the total production in the country.

Natural gas fields have been found in various regions of the country, the most widely known being the Shebelinskoe, Dashavskoe and Gazliyskoe. Of late, unique gas fields have been found and are being exploited intensively in Turkmenistan, South Urals and Tyumen district (Shatlykskoe, Orenburgskoe, Medvezhye, Urengoiskoe, Yamburgskoe). The gas reserves in these fields constitute almost 50% of the total explored reserves of natural gas in the country. Large reserves of gas and petroleum have been discovered in Komi autonomous district.



FUEL PREPARATION AT POWER STATIONS

3.1. Methods of Solid Fuel Combustion

Solid fuel combustion in boiler furnaces can be effected by various methods: flame combustion, cyclone combustion, or fluidized-bed combustion (Fig. 3.1). Flame combustion is the most popular in modern power engineering.

The classification of combustion methods is based on the aerodynamic characteristic of the process which determines the conditions of contact of the burning fuel with an oxidant [58].

The capacity of a furnace can be actually increased without limit by burning pulverized coal (coal dust) in a

suspended state in the furnace space. This is what is called flame combustion (Fig. 3.1a). In this method, fine particles of fuel are easily moved by the flow of air and combustion products through the section of the furnace. Combustion takes place in the furnace space within a rather short time of the presence of the particles in the furnace (1-2 s). The rate of fuel burning, and therefore, the amount of heat evolved in time, depend on the combustion surface.

In cyclone combustion, fuel particles go through intensive turbulent motion (Fig. 3.1b). In contrast to flame combustion, the fuel particles are blown

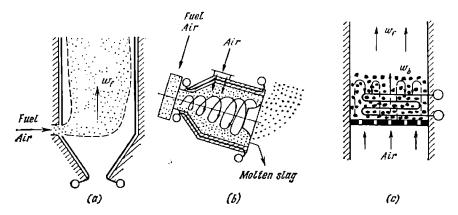


Fig. 3.1. Diagrams of combustion of solid fuels
(a) flame combustion; (b) cyclone combustion; (c) fluidized-bed combustion

over by the flow and burn off more quickly. The cyclone method permits the combustion of coarse coal dust and even crushed coal. A cyclone furnace develops a higher temperature with the result that the slags are removed in the molten state (slagging-type furnace).

In recent times, a new combustion method has come into use in the field of power engineering called fluidizedbed combustion (Fig. 3.1c). Solid fuel, ground to a particle size of 1-6 mm. is placed onto a grate and blown from beneath with an air flow at such speed that the fuel particles are lifted above the grate and are reciprocated in the vertical plane. In this process, the speed of the gas-air flow within the fluidized bed is higher than above it. The finer and partially burned particles rise into the upper portion of the fluidized bed where the flow velocity decreases, and there they burn completely. In operation, the fluidized bed increases in volume by 1.5-2 times; its thickness is usually 0.5-1 m.

Heat absorbing surfaces in the form of in-line or staggered tube bundles are arranged in and above the volume of the fluidized bed. The unit heat absorption within the fluidized bed increases substantially due to the intensive conductive (contact) heat tran-

sfer from incandescent fuel particles to heat-absorbing surfaces, though the temperature of gases in the burning bed remains at a relatively low level (800-1 000°C), which prevents the metal from overheating and diminishes the concentration of harmful nitrogen oxides in combustion products. In addition, this method of combustion makes it possible to introduce solid additions (say, limestone) into the bed in order to neutralize the sulphur oxides that form on combustion.

Large power stations consume as much as 1 000 t coal per hour or more. Even when delivered in cars of a large load-carrying capacity (60-125 t), 15-30 cars of fuel must be unloaded every hour, which is only possible with the use of highly efficient car dumpers.

The process of pulverization, i.e. transformation of lumpy fuel into dust, includes two stages (see Sec. 1.3). In the first stage, raw fuel is crushed to a particle size of not more than 15-25 mm. The crushed fuel is then delivered into raw coal bunkers and transferred to grinding mills where it is ground to a final particle size of no greater than 500 µm. During grinding, the fuel is dried by hot air to ensure it has good dust fluidity.

3.2. Pulverization Systems

The pulverization system is a combination of equipment in which solid fuel is ground, dried and tansferred to the burners of a boiler furnace.

By the method of delivery of pulverized fuel to the furnaces, pulverization systems can be divided into central and individual. In the former case, the system is arranged in a separate building (central coal pulverizing plant) where coal is pulverized on a controlized basis and then distributed through pipelines between the boilers of the station. In the latter case, each boiler is provided with its own pulverizing equipment, with certain provisions being made to transfer the pulverized fuel to neighbouring boilers so as to increase the reliability of the fuel supply.

The selection of a particular pulverization system for a power station presents a complicated technico-economical problem. Centralized pulverization systems turn out to be

Fig. 3.2. Individual pulverization system with direct blowing of pulverized fuel into furnace for the operation on compressed hot air

I—raw coal bunker; 2—cut-off gate valve; 3—coal feeder; 4 raw coal chute; 5—coal mill; 6—dust separator; 7—dust duct; 8—burner; 9—steam holler; 10 forced-draft fan; 11—afr henter; 12—primary nir path; 13—secondary alr path; 14—secondary air duct; 15—cold air for mill ventilation; 16—explosion relief valve; 17—flapper valve; 18—automatic cut-off gate valve

more efficient economically, especially when moist brown coal is pulverized, hut their equipment is more intricate and expensive, and, in addition to this, they are not sufficiently reliable in operation. Individual systems are simpler and more reliable and are widely employed by power stations [29].

Individual pulverization systems can be subdivided into the following types: closed systems which directly blow dust into the furnace space, closed systems with an intermediate dust bunker, open systems which use hot air as a dust carrier. The type of system (either closed or open) is determined by the way in which the drying agent is utilized upon fuel drying. In a closed system, it is directed into the furnace together with dried pulverized fuel; in an open system, the drying agent is carefully cleaned from fuel fines and ejected into the stack, bypassing the boiler furnace.

The pulverization system with closed fuel drying and direct dust blowing into the furnace. Crushed fuel is delivered from the coal hunker by a coal feeder into a grinding mill (Fig. 3.2). Hot air at a temperature $t_{ha} = 250-400$ °C is also fed into the mill in order to dry the fuel and transfer it further to the boiler furnace burners. This is what is called primary air. Coarse fractions of pulverized fuel are separated in a separator, after which the fuel and air (which has been moistened by the moisture of fuel) are supplied at a temperature of 80-130°C through pulverized fuel pipelines to the furnace burners. The remaining hot air, which is called secondary air, is fed separately to the burners.

The quantity of primary air to be used for the drying and transport of pulverized fuel depends on the fuel quality, in particular, the moisture content. The amount of primary air (r_1) is usually equal to 0.3-0.5 of the total consumption of air for combustion and increases with the moisture content. With extremely moist fuel,

the use of hot air only for fuel drying becomes economically inefficient and fuel combustion becomes unstable since the greater portion of pulverized fuel enters the combustion zone at a reduced temperature. In such a case, fuel is dried by a higher-temperature agent, say, by mixing the primary air with a part of the furnace gases.

If the pulverization system is rigidly connected with the boiler, it should satisfy more stringent requirements as regards the reliability of its operation. The number of installed grinding mills must be no less than three and the number of mills in operation minus one mill must ensure at least 90% of the rated load of the boiler. Hence the productivity of a mill, B_m , should be:

$$B_m \gg \frac{0.9B_b}{z_m - 1} \tag{3.1}$$

where B_b is the fuel consumption by the boiler at the rated load, kg/s, and z_m is the number of installed mills belonging to the boiler.

Upon exiting from the mill separator, fuel dust is divided between 2-4 pulverized fuel pipelines which are connected with different, not adjacent, burners. This is done in order to avoid an uneven temperature distribution in the furnace space should the mill be stopped for repairs.

In the scheme of Fig. 3.2 the resistance in the pulverized fuel path from the mill to the burners is overcome by the head developed by the forceddraft fan, so that the pulverization system operates under a slightly excessive pressure (supercharged); the pressure upstream of the mill is 1-2.5 kPa. An essential condition for the safe operation of the pulverization system and for maintaining the required cleanliness in the room is that the equipment must be kept perfectly tight.

The direct blowing system has certain advantages: it is simple, the pulverizing equipment is compact, tho consumption of electric energy for pust transport is low, and the fuel supply can easily be automatically controlled.

The pulverization system with closed fuel drying and intermediate dust bunker (Fig. 3.3). A characteristic feature of this system is that the prepared pulverized fuel is separated from the transporting air in a cyclone. The dust is directed into an intermediate bunker from which it is fed by special feeders into pulverized fuel pipelines. The moistened air at the exit from the cyclone has a temperature of 80-100°C and contains 10-15% of the finest coal dust. This air cannot be discharged through the stack and for this reason it is blown by the mill exhauster into the primary air duct to be distributed among the pulverized fuel pipelines (Fig. 3.3a). The number of pulverized-fuel pipelines and dust feeders is equal to the number of burners in the boiler furnace.

Due to the provision of the intermediate bunker, there is no need to match the productivity of the mill with that of the boiler, thus each of them can operate at the optimal load. The mill exhauster forms a negative pressure in the system, which avoids dust ejection to the surroundings. Air inleakage through places where the system is in contact with the surrounding atmosphere (raw coal chutes. dust chutes downstream of the cyclone) is prevented by means of flapper valves which open only for a short time to allow the mass of fuel that has accumulated on a valve to pass through.

In the combustion of low-active fuels with a low yield of volatiles, the temperature of the pulverized fuel air mixture should be raised to facilitate the ignition of the dust. This is achieved in a system where hot air is supplied together with pulverized fuel (Fig. 3.3b). In addition to the primary air which is fed into the pulverization system in an amount of 15-25%, another portion of hot air (20-25%) is directed into the air duct and then into the pulverized fuel pipelines by an auxiliary hot blast fan. In this case, the temperature of the

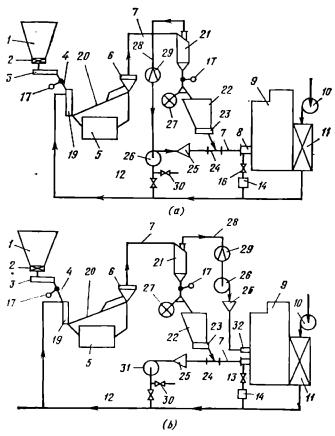


Fig. 3.3. Individual closed pulverization system with intermediate dust bunker (a) with pulverized fuel carried by drying agent; (b) with pulverized fuel carried by hot air and with drying agent discharged into boiler furnace. Items 1-17 as in Fig. 3.2; additional items: 19—fuel-drying device; 80—chute for return of coarse fractions; 21—cyclone; 88—pulverized fuel bunker; 23—pulverized fuel feeder; 24—mixer; 25—primary air duct; 28—mill ventilator; 27—reversible screw feeder; 28—moist drying agent with fuel fines; 29—flow meter; 30—valve for admitting cold air; 31—hot blast fan; 32—discharge burner

pulverized fuel air mixture is close to that of the hot air. The quantity of air supplied to the burners turns out, however, to be insufficient for complete fuel combustion. To correct this, low-temperature moistened primary air with a slight concentration of fine coal dust is fed from the cyclone into either the combustion zone through special discharge burners or into the annular channel around the main burners.

In the system described above, the load of the steam boiler is controlled by the dust feeders by using the reserve of pulverized fuel in the bunker. Usually two such systems are provided for a boiler. Their productivity in terms of fuel is 15-20% higher than the maximum fuel consumption by the boiler, because of which one of the systems remains inoperative for a certain time. The system can transfer part of the prepared pulverized fuel into the bunkers of other pulverizing systems through a reversible screw feeder. The available reserve of pulverized fuel in the bunkers allows short-term stoppage of both mills for inspection or repairs.

A disadvantage of the intermediate bunker system is that its equipment is too intricate and bulky. Furthermore, the system has an elevated hydraulic resistance, which increases the consumption of electric energy for dust transport. The storage of a large mass of dry dust increases fire and explosion hazard. Nonetheless, the system can reliably supply steam boilers with pulverized fuel and for that reason has found wide application.

The above-mentioned drawbacks of this system become especially pronounced in the operation of modern high-capacity boilers. In recent times, a new system of pulverized coal supply has been developed which is characterized by a high concentration of dust in fuel pipelines. In conventional systems, the concentration of dust in the primary air flow is usually 0.4-0.6 kg per kg air. In the new methpulverized fuel is transferred by compressed air, with a low air flow rate (only 0.1-0.3% of the total air flow rate to burners) and with a dust concentration as high as 30-60 kg/kg air. Since the quantity of air is not high, the dust acquires a high fluidity for motion through smalldiameter pipelines (60-90 mm). In burners, the dust is spread by hot air. The system does not require the bulky, 300-500 mm pipelines for the transport of dust from the dust bunkers to the furnace burners, sharply decreases the unit energy consumption

for pneumatic transport and enables one to adjust the flow rate of primary air to the boiler load, which is an improvement over the earlier systems.

Pulverization with the open drying system. Open pulverization systems are employed only with fuels which havo a resolved moisture content W^r of more than 3.6% kg/MJ. The extra cost and more intricate design of the system are compensated for by the higher efficiency of the boiler due to smaller heat losses with waste gases and lower unburned carbon loss in the furnace. A fuel with a high moisture content is dried by a high-temperature agent, i.e. combustion products at a temperature of 400-450°C, which are taken off in an amount of 6-10% of the gas volume from the gas duct downstream of the economizer (Fig. 3.4). In another version, drying is effected by a mixture of hot air and combustion products, with the mixture temperature being 500-600°C.

The worked-off drying agent from the cyclone is fed, together with the unseparated finest fuel fractions (roughly 10%), into the second stage of dust collection, where coal dust is separated in multicyclones (a set of 150-250 cyclone elements of small diameter), electrostatic precipitators or cloth bag filters. The separated dust flows by gravity through chutes into

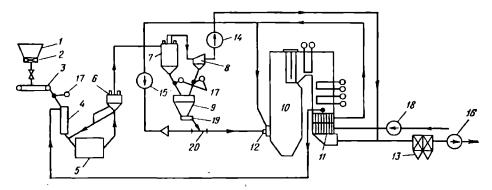


Fig. 3.4. Open pulverization system with gas drying of fuel

1- raw fuel bunker; 2-cut-off gate valve; 3-raw coal feeder; 4-gas and raw coal mixer; 5-grinding mill-6- dust separator; 7-cyclone; 8-dust collector; 9-dust bunker; 10-steam holler; 11-air heater; 12-burner; 13-main electrostatic precipitators; 14-mill ventilator; 15-bot blast fan; 16-induced-draft fan; 17-flapper valve; 18-forced-draft fan; 19-pulverized fuel feeder; 20-pulverized fuel-air mixer

an intermediate dust bunker, while the drying agent after the dust collectors is discharged into the main electrostatic precipitators of the boiler plant and is then combined with waste gases. The transport of the drying agent is effected by the mill exhauster, and the supply of primary air by the hot blower (hot blast fan).

The open system of fuel drying substantially improves the quality of fuel and increases the efficiency of fuel combustion. The volume of combustion products in the boiler flue ducts diminishes, which results in lower acrodynamic resistance and a lower waste gas temperature.

An essential drawback of the open system is that some fuel fractions are lost with the discharged drying agent. Another drawback is an elevated energy consumption for separation and purification of the moist drying agent. Despite the complicated system of dust collectors, roughly 1-2% of the fuel are lost to combustion and discharged into the atmosphere, resulting in air pollution. The loss of fuel may rise substantially with improper operation of the dust collectors or a high moisture content in the drying agent. For these reasons, application of the open system is limited only to cases of rather moist fuels which cannot be burned efficiently by conventional methods.

3.3. Characteristics of Coal Dust. Optimal Degree of Pulverization

Upon being ground in a mill, pulverized coal is essentially a polydisperse powder with a particle size of up to 300-500 μm. Coarse-ground brown coal may have a particle size of up to 1 mm. Fine particles of pulverized coal can adsorb an appreciable quantity of air on their surface. Freshly poured coal dust has a density of $\rho = 500-600 \text{ kg/m}^3 \text{ but is slumped in}$ the course of time and its density increases to 800-900 kg/m³. When mixed with air, coal dust has a high fluidity

and can be easily transported in pipe-

Sieve analysis of pulverized coal. The quality of pulverized coal is determined by the milling fineness (milling dispersity) and the relative concentration of individual fractions. These characteristics are determined by sieve analysis. A sample of pulverized coal is screened through 4-5 sieves with a progressively decreasing mesh size. In the Soviet Union sieves are characterized by the clear size of mesh expressed in micrometres. Using the standard technique of sieve analysis, the total sieve residue is determined by the total number of particles of a size greater than the given mesh size x, μ m, expressed as a percentage of the initial sample mass. This residue is designated as R_x . It includes the dust residue on a given sieve and that on all other sieves above it with larger mesh sizes.

Sieves can be also used for the fractional separation of dust with a particle size of not less than 40 µm. Finer-fraction particles stick readily to the sieve material and coalesce with one another. For this reason. the finest dust is blown in an air classifier to grade it to size.

For more clarity and convenience, the results of a sieve dust analysis can be represented graphically as a particle-size distribution curve. If the mesh sizes of sieves are laid off as the abscissae, x, and the total sieve residues, R_x , as ordinates, the resulting curve is called the integral particle-size distribution curve, or total-residue curve (Fig. 3.5). The curve is plotted by using the results of screening through sieves and of blowing the finest fractions in an air classifier. This method of dust size analysis is, however, labour-consuming and takes up much

The analysis of numerous particlesize distribution curves for various kinds of fuel has demonstrated that all such curves can be described by the equation:

$$R_{\mathbf{x}} = 100e^{-tx^n} \tag{3.2}$$

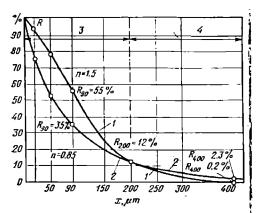


Fig. 3.5. Complete grain-size characteristics of pulverized brown coal obtained in mills of two types

I—grinding in hammer mill; 2—grinding in hall mill; 3—region of fine dust fractions; 4—region of coarse dust particles

where b and n are constants for a given fuel and particular grinding method. They turn out to be unknown on grinding fresh dust (they can be determined by screening a sample of fuel through two sieves, with the mesh size $x = 90 \mu m$ and $200 \mu m$).

Of particular importance is the coefficient n which is called the *polydispersity coefficient of dust*. It characterizes the structure of dust and its particle-size distribution. This can be seen upon differentiation of equation (3.2) with respect to x:

$$y = -\frac{dR_x}{dx} = 100bnx^{n-1}e^{-bx^n} = R_xbnx^{n-1}$$
(3.3)

In this case, the ordinate $y, \%/\mu m$, in Fig. 3.6 will give the concentration, %, of particles of the size x. With n > 1, the curve has a maximum in the zone $x = 15\text{-}25 \mu m$. Such dust has a relatively small content of the finest fractions. With n = 1 and n < 1, on the contrary, the highest quantity of particles falls on the very fine fractions. The loss of heat with unburned carbon in boiler furnaces depends mainly on the concentration of coarse particles in pulverized coal (of a size more than 250 μm).

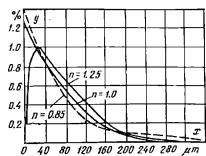


Fig. 3.6. Differential curves of the fractional distribution of pulverized fuel particles

Energy consumption for pulverization. When grinding a particular material, it is possible to establish certain quantities which will characterize the energy consumption in grinding. As relates to grinding fuels in grinding mills, Rittinger's law is commonly used and is formulated as follows: the work spent on grinding a material is proportional to the newly formed surface area:

$$E = \frac{N}{H} = A (f_d - f_c)$$
 (3.4)

where E is the energy of grinding, kW h/kg, N is the mill power, kW, B is the mill productivity, kg/h, f_c and f_d are the initial surface areas of 1 kg crushed fuel and the final surface area of 1 kg of produced dust, respectively, m^2/kg , and A is the unit consumption of electric energy in grinding related to 1 m^2 of newly formed surface, kW h/ m^2 .

Since f_d is much larger than f_c , expression (3.4) can be simplified:

$$E \approx A I_d$$
 (3.5)

The unit consumption of energy A has been measured for most kinds of fuel, and therefore, it is possible to determine the energy consumed in fu-

el grinding if the surface area of the dust is known.

The surface of dust. As illustrated in Figure 3.6, the theoretical surface area of pulverized fuel consisting of differently sized particles can be found with the following simplified formula—assuming the particles are spherical or cubic in shape and have the polydispersity coefficient n:

$$f_{th} = \frac{450 \times 10^3}{P_f} \frac{1}{n} \left(\ln \frac{100}{R_{10}} \right)^{1/n}$$
 (3.6)

where ρ_f is the unit density of ground fuel, kg/m^3 , usually $\rho_f = 1700-1840 \text{ kg/m}^3$.

The actual surface area can be found by introducing the shape factor k_s into the formula above. The average value of the shape factor for pulverized coal may be taken as $k_s = 1.75$. The actual surface area is then found as $f_d = k_s f_{th}$.

The surface area of the dust is determined mainly by the concentration of fine fractions. The loss of heat with unburned carbon in chamber furnaces is usually associated with a high content of coarse fractions (above 250-300 μ m for anthracites and coals with the yield of volatiles V^c less than 25% and above 500-600 μ m at V^c more than 30%). It then follows that pulverized fuel should have n > 1, a small concentration of fine fractions (below 25 μ m) and a small concentration of coarse fractions.

Moisture content of pulverized fuel. Moisture content W, %, is an important characteristic of pulverized fuel. An increase in moisture content above the recommended level may result in lower boiler productivity and involve difficulties in dust transport: the dust loses fluidity and slumps in bunkers, clogs feeders, chutes, etc. On the other hand, overdried coal and brown coal dust is liable to self-ignite in places where it is stored or accumulates and may be explosive when mixed with air. The moisture content of pulverized fuel is usually determined in terms of the hygroscopic moisture content W^h (see Sec. 2.3).

Dust explosiveness. When coal dust suspended in air is confined in a closed volume, it will explode more intensively if its unit surface area is larger (i.e. if it contains more fine fractions) and if it has a higher yield of volatiles. The temperature of the mixture is also of crucial importance. The most dangerous concentrations of coal dust lie within 0.3-0.6 kg/m³ air.

The ignition of an air-dust mixture in a closed volume results in a sharp rise in temperature and pressure. The pressure may rise well above the limits safe for pulverizing equipment. Damage to the boiler plant by explosions is prevented by installing safety (relief) valves which discharge part of the mixture from the system should the pressure rise excessively. The concentration limit of O₂ in the drying agent, i.e. the concentration below which fuel dust cannot explode, is equal to 16% for pulverized peat and oil shales, 18% for brown coals, and 19% for coals. The concentration of O, can be decreased by drying pulverized fuel with a mixture of hot air and combustion products. The probability of explosion is lower in fuels with a lower yield of volatiles. With the yield of volatiles less than 8%, fuel is explosion-safe. A high temperature of the dust-air flow promotes the formation of explosive mixtures, and therefore the temperature of the air-dust mixture downstream of the mill must be strictly controlled. For most pulverized fuels, it should not exceed 70-80°C (fuels with a high yield of volatiles) and 130°C in other cases.

The optimal degree of grinding. The grinding properties of various fuels are compared in terms of the laboratory coefficient of grindability, k_l , which is understood to be the ratio of the unit consumptions of electric energy for grinding, in a standard laboratory mill, a reference solid fuel and the fuel being tested, provided that both have the same initial particle size and the same ground dust characteristics. Thus

$$k_1 = E_t / E_t \tag{3.7}$$

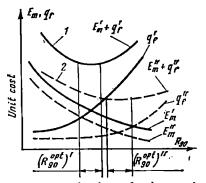


Fig. 3.7. Determination of the optimal grinding degree of fuel

1—fuel with low V^c and low h_i: 2—fuel with higher values of V^c and h_i

The conditions in industrial grinding mills may differ from the standard laboratory conditions in the initial moisture content of fuel and its particle size, therefore, the grindability k of working fuel may differ from the laboratory value k_l .

As follows from formula (3.4), grinding of fuel to a coarser size results in energy savings, ΔE_{pul} , but inevitably increases the time of complete combustion in the furnace and the heat loss with unburned carbon ΔB_{ℓ} . Therefore, each kind of fuel has a particular fineness range within which the total expenditures on mill grinding E_m and the cost of heat loss with unburned carbon in the furnace, q_t , will be at a minimum. Expressing these items in the same units, say, in roubles per ton dust, they may be plotted and compared in a single graph (Fig. 3.7). The fineness of grinding is usually estimated in terms of the sieve residue R_{90} . To find the optimal value

of R_{00} , the boiler plant and the pulverization system are tested at different degrees of fuel grinding.

The optimal value of R_{90} depends even more on the yield of volatiles V^c , type of mill, and type of dust separator. The effect of these factors is expressed by the polydispersity coefficient n. The optimal fineness of fuel grinding can be found by the formula:

$$R_{cpt}^{opt} = 4 + 0.8nV^{c}$$
 (3.8)

3.4. Pulverization Equipment

The grinding mill is the principal element in any pulverization system. The characteristics of the main types of grinding mills are given in Table 3.1. They differ from one another in the principle of fuel grinding and the rotational speed of the movable element. Ball-tube mills and hammer mills are used most often for the purpose, the former being preferred for grinding fuels with n relatively low yield of volatiles and the latter for grinding younger coals, brown coals, peat, and oil shales. Around 98% of the total fuel ground in pulverization systems fall on these two types of mill. Some kinds of coal can be ground more efficiently in roller mills. 'Soft' (with $k_1 > 1.5$) and very moist brown coals are ground in pulverizing fans.

A ball-tube nitl (or simply ball mill) has a drum 2-4 m in diameter and 3-10 m long which is partially filled with steel balls 30-60 mm in diameter (Fig. 3.8). The drum is clad with armour plates on the inside and is heat- and sound-insulated on the outside. Raw fuel and hot air are supplied to the drum through the inlet.

Table 3.1. Grinding Mills Characteristics

Mill type	Grinding principle	Rotational speed, s-1 (rpm)	Speed characteris- tic
Ball-tube mill Roller mill Hammer mill Paddle-type mill Pulverizing fan	Impact, abrasion	0.25-0.42 (15-25)	Low-speed
	Crushing	0.85-1.3 (50-80)	Medium-speed
	Impact	12.5-16.3 (750-980)	Iligh-speed
	Impact	25 (1500)	Iligh-speed
	Impact	12-24.5 (735-1470)	High-speed

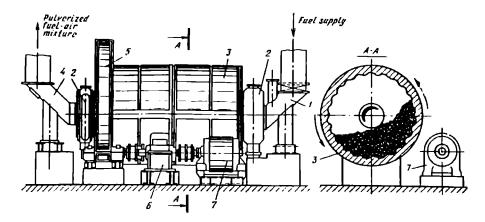


Fig. 3.8. Ball mill (general view and cross section)

1-inlet pipe; 2-supporting bearing; 5-heat- and sound-insulated mill drum; 4-outlet pipe; 5-harger gear wheel; 6-reducer gear; 7-electric motor

pipe. The drum is rotated by an electric motor via a reducer gear and a driven gear wheel attached to the drum.

The optimal capacity of a mill is obtained at a rotational speed

$$n_d = 0.76n_{er} (3.9)$$

where n_{cr} is the critical rotational speed of the drum, s^{-1} , at which balls can 'stick' to drum walls due to centrifugal forces:

$$n_{er} = \frac{0.705}{V \overline{D_d}} \tag{3.10}$$

where D_d is the drum diameter, m. With the optimal rotational speed of the drum, the balls first rise over the drum wall and then detach from it and fall. Fuel is ground by the impact of the falling balls and by the abrasion between them. Final dust is continuously removed from the mill by the ventilating agent—air.

The grinding capacity of a mill, B_m , depends substantially on the drum length and diameter. Yet a larger diameter D_d appreciably increases the required power of mill motor N_m which is roughly proportional to the third power of D_d .

Grinding mills are also characterized by the drying capacity, i.e. the quantity of fuel that can be dried in

the mill from the original moisture content W^w to the desired value W^d . The two characteristics of a mill should be matched properly, which is dene by controlling the flow rate and temperature of the drying agent at the mill inlet.

The power for mill rotation N_m is virtually independent of the mass of charged fuel in view of the large mass of balls and drum. For this reason, as the quantity of charged fuel decreases, the unit energy consumption for grinding, E_g , kW h/kg, increases, since

$$E_{g} = N_{m}/B_{m} \tag{3.11}$$

Thus, it is advisable to run ball mills at full load.

The armour plates and balls of a mill are inevitably subject to wear during mill operation. The extent of wear depends on the abrasive properties of fuel which can be characterized by the relative abrasivity k_{abr} , which is the ratio of the actual wear to that taken as the reference value $\{0.3\ g/(kW\ h)\}$.

As has been found by experiments, the two relative characteristics, k_I and k_{abr} , are closely interrelated (Fig. 3.9). Harder fuels cause greater abrasion wear of the mill elements [29].

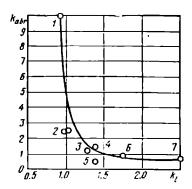


Fig. 3.9. Dependence of k_{abr} on k_1 s = Donelsk anthracite roat; z= Donelsk gas coat; z= Chelyabinsk coat grade B; t= Vorkuta coat grade GZh; s- Karaganda coat grade GZh; s= Moscow district coat grade B; t- Estonian offishales

The wear of the balls is compensated for by adding new balls periodically into the operating mill. Ball-tube mills are suitable for grinding all kinds of coal. Metallic objects occasionally present in coal are not dangerous to them. On the other hand, ball mills, as a rule, make coal dust with the polydispersity coefficient n less than unity. For this reason and also because of their large dimensions, they have a typically high unit consumption of energy (15-25 kW h/t for coals and up to 35 kW h/t for anthracites). The application of ball mills in pulverization systems with an intermediate bunker is profitable only when grinding high-abrasive low-reactive fuels with $R_{10} \leq 10\%$ and $k_i \leq 1.1$.

A hammer mill has a steel casing clad on the inside with smooth armour plates 20-30 mm thick and a rotor with discs to which hammer holders and hammers are hingedly attached. In operation, the circumferential speed of the hammers attains 50-60 m/s. Rotating hammers break fuel lumps into smaller pieces which then strike the armour and are additionally disintegrated by abrasion in the gap between the hammers and casing. The hammer mill is usually installed together with a dust separator (Fig. 3.10).

The basic parameter that determines the effectiveness of grinding is the circumferential speed of the hammers. The productivity of a mill is proportional to the cube of circumferential speed. The productivity of mills can be increased by increasing the rotational speed or the rotor diameter, but this increases the dynamic load on the rotor, resulting in a heavier mass of the mill and higher energy consumption. As the coal charge of a mill increases, the energy consumed increases first relatively slowly and then rather quickly. The unit energy consumption for grinding [see formula (3.11) decreases smoothly to a minimum at a certain charge of the mill, making it possible to vary the mill

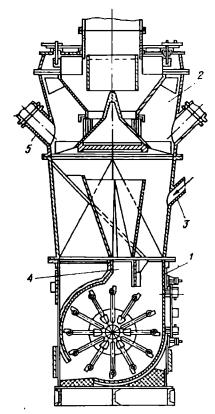


Fig. 3.10. Hammer mill with centrifugal dust separator

f—hammer mill; z—dust separator; 3—raw fuel inlet; 4—chute for return of coarse dust; 5—safety valve

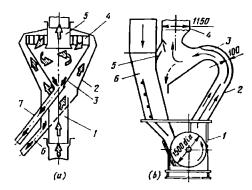


Fig. 3.11. Dust separators

(a) centrifugal dust separator for coals and anthra cites; 1-pipe for admitting dust-air mixture 2-external bousing; 3-internal housing; 4-swirl vanes; 5-dust-discharging pipe; 6-chute for the return of coarse dust from external housing; 7-ditto, from internal housing; (b) inertial separator for brown coal and coals; 1-hammer mill; 2-separator housing; 3-hag for collecting coarse fuel particles; 4-dust-discharging pipe; 5-controlling gate valves; 6-chute for delivery of fuel and drying agent into mill

charge in a rather wide range without noticeably worsening the mill efficiency. For that reason and in contrast to ball mills, hammer mills can participate in controlling the boiler load.

Hammer mills may be recommended for grinding fuels with $k_l \ge 1.1$ which can be burned as a rather coarse dust (brown coals, coals with the yield of volatiles $V_c > 28\%$, peat and oil shales). With such fuels, the unit energy consumption of hammer mills is 33-50% lower than that of ball mills (8-12 kW h/t).

Roller mills are usually employed in systems with direct dust blowing for grinding moderately hard coals with relatively low humidity and a low content of hard fractions (pyrite) in the mineral portion, such as Ekibastuz coals. Series manufacture of roller mills has recently been started in the USSR.

A common disadvantage of all reller mills is that they are sensitive to metallic objects—should these get occasionally into the mill with coal. Other drawbacks include uneven wear of the working members and certain repair complexities. On the other hand, roller mills are advantageous in having a low unit energy consumption (12-15 kW h/t), small dimensions, and a low noise level.

Soft brown coals $(k_1 > 1.5)$ of high humidity are pulverized in pulverizing fans. The rotor of a fan carries a pulverizing wheel with hammer blades arranged at its periphery. The grinding effect is produced by the impact from rapidly rotating hammer blades on fuel lumps. The blades create a head (1 200-1 500 Pa) behind the fan, which obviates the need for a mill exhauster. Pulverizing fans are noted for their simple design, small dimensions and low cost. Their productivity is, however, not high, ranging from 3.5 to 45 t/h for brown coal (at $R_{90} = 55\%$ and $k_1 = 1.7$).

The auxiliary equipment of pulverization systems includes dust separators, cyclones, raw coal feeders, dust feeders, and bunkers.

A dust separator is used for controlling the fineness of fuel dust produced in a mill. The separation of dust into coarse and fine fractions in dust separators is effected by using centrifugal, inertial and gravitational forces. The effectiveness of operation of a separator determines to a large extent the quality of the dust supplied into boiler furnace burners. The type of separator is usually selected according to the characteristics of the respective mill. Centrifugal and inertial dust separators are the most popular (Fig. 3.11).

separator centrifugal 3.11a) has two cones inserted one into the other. The dust-air flow is supplied into the separator bottom at a flow rate of 15-20 m/s. Owing to expansion, the flow rate in the annular space between the cones drops down to 5-6 m/s, producing the effect of gravitational separation. The coarsest particles slide down along the walls of the cone and return to the mill. Tilting vanes arranged at the inlet to the inner cone cause the flow to swirl, producing the centrifugal effect. The necessary fineness of dust is obtained by varying the angle of the vanes.

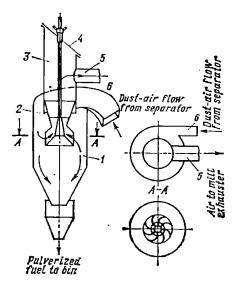


Fig. 3.12. Cyclone dust separator

1- housing: 2-control pipe with vanes: 3-internal cylinder: 4-explosion relief valve; 5-exit
of transfer agent (air); 6-inlet pipe

In an inertial separator (Fig. 3.11b), dust separation into fractions is brought about by changing the direction of dust-air flow, i.e. by utilizing the forces of inertia. The magnitude of ΔR_{00} can be controlled within 10-15% by varying the position of a gate valve. Inertial-type separators are employed in combination with hammer mills for coarse pulverization of brown coals, oil shales and milled peat (with $R_{00} > 40\%$).

Cyclones are employed in systems with intermediate bunkers for separation of pulverized fuel from transporting air (Fig. 3.12). The separation takes place as a result of the contrifugal effect and the deflection of the flow into the central outlet pipe. The efficiency of cyclones is 80-93% and increases inversely with the diameter.

Raw coal feeders are used for uniformly controlling the supply of coal to the grinding mills. The productivity of a feeder is selected with a reserve of 20% above the rated mill productivity. At power stations, belt feeders (for dry coals) and chain feeders (for

moist coals prone to smearing) are most often employed.

Dust feeders are arranged under dust bunkers and are used to control the dust supply into the pipelines. Screw-type and paddle dust feeders are the most common at power stations. The throughput capacity of a feeder is controlled by changing the rotational speed of d.c. electric drive motors.

Dust bunkers are reservoirs for the storage of a certain reserve of pulverized fuel. The volume of a bunker is determined so as to ensure the operation of the boiler at the rated load for 2-3 hours upon stoppage of the pulverization system. To evenly discharge dust into the dust feeders, the height of the dust bed in a bunker should be no less than 3 m.

3.5. The Preparation of Fuel Oil and Natural Gas

To facilitate transport, filling and discharge from reservoirs, fuel oil should have a temperature of at least 60-70°C, at which its viscosity will be not more than 30° E.

The flow path of fuel oil preparation at a power station (Fig. 3.13) comprises a receiving-discharging facility (pouring trestles with chutes and receiving vessels with submerged pumps), main reservoirs for storing a permanent reserve of fuel oil, fuel oil pumping house, fuel oil and steam pipelines, fuel oil heaters, and filters. The preparation of fuel oil for combustion consists in separating mechanical impurities and increasing the pressure and temperature of fuel oil. This is essential for minimizing the energy consumption of transporting fuel oil to steam boilers and for its fine atomization in burners. The tomperature of fuel oil when in tanks is maintained at a level of 60-80°C in all seasons by circulation preheating, i.e. by recirculating back into the storage tank a part (up to 50%) of the fuel oil that has been preheated in external hoaters.

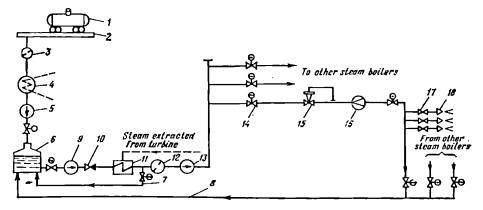


Fig. 3.13. Flow diagram of fuel oil preparation at a power station 1—fuel oil tank car; 2—draining device; 3—coarse filter; 4—heated discharge tank; 5—pump; 6—main tank; 7, 8—fuel oil recirculation lines; 9—first-lift pump; 10—back-pressure valve; 11—fuel oil heater; 12—fine filter; 13—second-lift pump; 14—closing gate valve; 15—flow rate controller; 16—flow meter; 17—gate valve; 18—hurner

A two-stage scheme of fuel oil supply is typical for thermal power stations. In this scheme, preheating the fuel oil, agitating it in tanks, and filtering are performed at a low fuel oil pressure (around 1 MPa), while second-lift pumps supply fuel oil to the main building at a high pressure (3.5-4.5 MPa).

The fuel oil is ejected from the atomizing burners at a high rate, which may result in the quick abrasive wear and failure of the burner channels. Also, small channels (less than 3 mm in diameter) may become clogged with large solid particles or blobs of tar or asphalt. Purification of fuel oil from solid impurities is performed first in coarse filters with a mesh size of 1.5 mm × 1.5 mm and then in fine filters with 0.3-0.5 mm meshes, both of which are mounted upstream of the pumps of hot fuel oil in the second-stage fuel oil line.

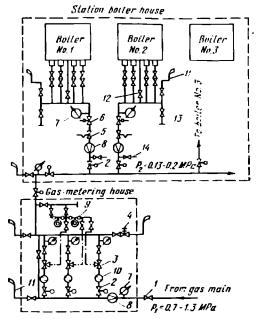
The fuel oil is preheated in steam heaters to a temperature no greater than the ignition point of fuel oil vapours. The fuel oil temperature is maintained at the specified level, irrespective of the fuel oil consumption by the boiler, by continuously recirculating the fuel oil through the storage tank.

Gaseous fuel is supplied to a power station from a gas main or gas-distribution station at a pressure of 0.7-1.3 MPa. Power stations are not provided with gas holders. The supplied gas is throttled in a gas-metering house until it reaches the pressure it should have in the burners (0.13-0.2 MPa). Since gas throttling produces much noise and involves a high explosion hazard, the gas-metering house is located in a separate building at the station site (Fig. 3.14).

Each gas-metering house has several (most often three) gas pipelines with pressure regulators, one of which is a stand-by line. A bypass line around the pressure regulators is also provided. The gas is purified from mechanical impurities by filters arranged upstream of the control valves. The control valves maintain the specified pressure 'downstream'. Safety valves are provided to protect the syagainst occasional pressure jumps. The quantity of gas passing through the gas-metering house is measured by flow meters. Each gas pipeline leading to a boiler has an automatic flow governor and quick-closing cut-off valve. The automatic flow governor adjusts the gas flow rate to the current load of the boiler. The pulse-

Fig. 3.14. Gas-supply scheme of power station

1—hand-controlled gate valve; 2—electric motor-driven gate valve; 3—gas pressure regulator; 4—safety valve; 5—gas flow rate regulator; 6—quick-action cut-off valve; 7—manometer; 8—flow meter; 9—pressure regulator sensors; 10—filter; 11—gas vent; 12—gas supply to burner igniter; 13—drain; 14—gas sampling for analysis



operating cut-off valve stops the gas supply to the boiler furnace in emorgency situations when the supplied gas might cause explosions (flame breakoff, loss of air pressure at the burners, shutdown of the motors of exhaust or forced-draft fans, etc.).

To avoid explosion hazard during standstills, the gas pipelines are blown through with air through special gas vents. These vents are mounted outside of the building in places that are inaccessible to people. Gas vents also serve to displace the gas-air mixture from gas pipelines before firing a boiler. The blow-through of a gas line is stopped when the concentration of oxygen in a sample is not more than 1%.

All gas-supply facilities of a power station should not be operated without observing the necessary safety precautions.



THEORETICAL PRINCIPLES OF COMBUSTION

4.1. The Kinetics of Combustion Reactions

The combustion of fuel in a furnace involves a number of complex physical and chemical processes. Among

the physical phenomena which precede or accompany the ignition and subsequent burning of the fuel, special emphasis should be laid on the process of mixing the fuel and the oxidant (aerodynamic factor). Among the chemical factors, the temperature and concentration of the reacting substances are the most essential. One also has to deal with the complex fields of velocities, concentrations and temperatures, which together determine the kinetics of chemical reactions.

The reactions involved in fuel combustion proceed with heat evolution, i.e. they are exothermic. These include the burning of carbon, hydrogen and sulphur in atmospheric air. At high temperatures, such as in the core of a flame, some reactions may occur with absorption. These are called endothermic reactions. The formation of nitrogen oxides $N_2 + O_2 = 2NO - 180$ kJ/mol and the reduction of carbon dioxide to monoxide on the incandescent surface of coke particles under conditions of oxygen deficiency $C + CO_2 = 2CO - 7.25 \text{ MJ/kg}$ are both examples of endothermic reactions. Chemical reactions which can proceed in either a forward or reverse direction are called reversible, though the conditions that foster the reaction in the forward or reverse direction may differ substantially from one another. For instance, the formation of nitrogen oxides can take place only at very high temperatures, while the reverse reaction, i.e. their dissociation into gaseous oxygen and nitrogen, can occur in the earth's atmosphere under the effect of solar radiation. During fuel combustion in furnaces, the rate of a direct process is immeasurably higher than that of its reverse counterpart, meaning that the equilibrium of these reactions is shifted towards the formation of the final products of the direct reactions, and therefore, these processes can be regarded as irreversible.

The intensity of combustion is characterized by the rate of the reaction involved. The rate of a homogeneous reaction is understood as the mass of a substance that reacts in unit volume per unit time. The reaction rate is measured in such a case in terms of the variation of the concentration of one of the reacting substances,

 ΔC_A , mol/m³ or g/m³, in a given time interval $\Delta \tau$, s:

$$\bar{w_r} = -\Delta C_A/\Delta \tau \qquad (4.1)$$

The rate of a heterogeneous reaction is expressed as the quantity of substance, Δg , g, that has reacted on the surface of solid combustible substance S, m^2 , during a time interval $\Delta \tau$, s:

$$\vec{K}_s = -\frac{\Delta c}{S\Delta \tau} \tag{4.2}$$

Reaction rates obey the law of mass action, according to which the rate of a reaction in a homogeneous medium at a constant temperature is proportional at any given moment to the product of the concentrations of the reagents:

$$w_r = kC_A^m C_B^n \qquad (4.3)$$

where k is the reaction rate constant which depends on the temperature and chemical nature of the reagents, C_A and C_B are the current concentrations of the reagents A and B in the gas mixture, and m and n are the number of moles of A and B, respectively.

Since the partial pressures of the components in a gas mixture at a constant temperature are proportional to their concentrations, formula (4.3) can be written in another form:

$$\boldsymbol{w}_r = k \boldsymbol{p}_A^{\mathsf{m}} \boldsymbol{p}_B^{\mathsf{n}} \tag{4.4}$$

In heterogeneous combustion, the concentration of the combustible substance (solid fuel) is constant, and therefore, the rate of this reaction depends only on the concentration of the oxidant (oxygen) on the surface of solid fuel:

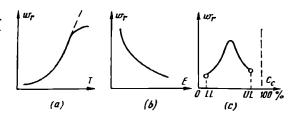
$$w_r = kC_{02}^s \tag{4.5}$$

where C_0^{σ} is the concentration of oxygen on the fuel surface.

If the concentrations of the reagents do not vary in time, the reaction rate is determined by the reaction rate constant k which depends on the temperature and nature of the substances that enter the reaction. This relationship is expressed by Arrhenius' law:

$$k = k_0 e^{-E/RT} \tag{4.6}$$

Fig. 4.1. Dependence of reaction rate on (a) temperature, (b) activation energy, and (c) concentration of combustible matter LL—lower limit; UL—upper limit



where k_0 is the pre-exponential factor, E is the activation energy, R is the gas constant, and T is the absolute temperature of the process.

Thus, the rate of a reaction quickly increases with temperature (Fig. 4.1a). A chemical reaction is possible due to the collisions of the molecules of reacting substances. If all collisions resulted in a reaction, combustion would occur at an enormous rate and the reaction rate constant would be equal to k_0 . Actually, a reaction can take place only when the energy of collisions can break the old molecular bonds and the fragments of old molecules can rearrange themselves into molecules of new substances.

The energy that is sufficient to destroy the molecular bonds of the starting substances is called the activation energy E. With a higher activation energy, the molecular bonds of the original molecules are destroyed less easily, and consequently, the reaction rate is lower (Fig. 4.1b). For instance, the activation energy of the reaction $C + O_2 = CO_2$ is E_{CO_2} = 140 kJ/mol and that of the reac- $C + \frac{1}{2}O_2 = CO_1 E_{CO}$ = 60 kJ/mol. It then follows that the rate of formation of carbon monoxide in the reactions of carbon oxidation by oxygen is substantially higher than that of the formation of carbon dioxide CO₂, and therefore, carbon monoxide CO will be formed predominantly at the surface of burning carbon [29, 48].

Combustion in furnaces requires the continuous supply of fuel and exident (air) to the combustion zone, and therefore, the concentrations of the reagents are practically invariable in time. Under such conditions, the hig-

hest reaction rate can be attained at a nearly stoichiometric ratio of the concentrations of reagents. An excess of fuel (rich mixture) or a fuel deficiency (lean mixture) will cause the rate of the reaction to decrease due to lower heat evolution per unit volume.

Combustion cannot take place at any arbitrary concentration of fuel in the mixture, but only in a definite range of its concentrations in the air (oxidant). There exists a lower concentration limit below which combustion is impossible, and an upper concentration limit when any further increase of the concentration of the fuel prevents combustion (Fig. 4.1c). Thus, combustion is possible only in the concentration range between these two limits.

If a gas-air mixture is prepared for combustion and completely fills in a particular volume, an ignition source will induce the oxidation reaction which will propagate with a high rate all over the volume, resulting in a sharp increase in temperature and pressure. This type of process is called explovise combustion, or simply explosion. In such cases, we speak about the upper and lower explosive limits of a mixture, meaning that a gasair mixture is capable of exploding in the whole range of concentrations between these limits. The temperature above which a self-sustained reaction is possible in a mixture (system) is called the *ignition temperature*.

The relationship between heat evolution and heat removal at various temperatures of a combustible mixture can be represented graphically (Fig. 4.2). Heat evolution at the initial stage of a chemical reaction can

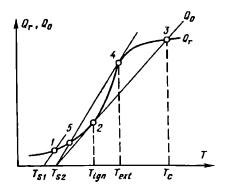


Fig. 4.2. Graphical determination of ignition temperature (T_{tg}) , combustion temperature (T_c) and extinction temperature (T_{ex}) at a given temperature of heat-transfer surface (T_e)

be found from the equation:

$$Q_r = k_0 e^{-E/RT} C_c^m V_m q \qquad (4.7)$$

The quantity of heat removed from the reaction zone is determined by the equation:

$$Q_0 = \alpha S (T - T_s) \qquad (4.8)$$

In these formulae, C_c is the concentration of the combustible substance, V_m is the volume of the prepared gas-air mixture, q is the thermal effect of the reaction per unit mass of fuel, α is the heat-transfer coefficient, S is the surface area of the cooling walls, and T and T_a are the temperature of the mixture and the cooling walls, respectively.

Heat evolution at the initial stage of reaction can be described by an exponential curve with Q_r rising substantially. In the region of high temperatures, the growth of Q_r is limited by the possibilities of oxidant supply, i.e. by the conditions of mixing. Assuming $\alpha = \text{constant}$, heat removal can be represented by a straight line inclined at an angle determined by α .

If the fuel-air mixture in a chamber and the surrounding walls are preheated to a temperature T_{si} , owing to heat evolution in the reaction, the mixture will be heated to a temperature $T_i > T_{si}$. In point I, $Q_r = Q_0$ and further preheating of the mixture is

impossible. The state in point 1 can be characterized by an insignificant heat evolution and low temperature. This is the zone of slow oxidation which may continue for a long time.

Let the temperature of the cooling walls be raised up to T.2. Again $Q_r > Q_0$ initially, and the temperature of the mixture rises. Finally, in point 2, $Q_r = Q_0$, but in contrast to point I, the state in this point is unstable. A slight rise of temperature is sufficient to make Q_r greater than Q_0 again and heat evolution will increase more quickly than heat removal. The temperature in point 2 is exactly what is called the ignition temperature T_{ign} . The process now develops rapidly to a state of stable hightemperature oxidation, or combustion (point 3). This process will be stable as long as the fuel and oxidant are supplied properly. If heat removal is done more forcibly (along a steeper line Q_0), the combustion temperature will decrease to point 4 where the high-temperature process will be interrupted since $oldsymbol{Q_0}$ becomes greater than Q_r in the whole temperature range up to point 5. The temperature of a combustible mixture in point 4 is called the extinction temperature. Combustion extinction always takes place at a higher temperature than ignition does, since the concentration of the starting substances in the zone of active combustion is lower than their initial concentration at ignition.

4.2. The Mechanisms of Fuel Combustion

As has been found by experiment, the rates of combustion reactions substantially exceed the rates calculated using the law of mass action and Arrhenius' law by considering the number of active molecules of the initial substances entering a reaction. Actually, reactions do not occur immediately between the original molecules, but pass through a number of intermediate stages in which active mole-

cular fragments (radicals and atoms of H. OH, O, etc.) participate along with molecules. As a result, each of these intermediate reactions has a low level of the activation energy E, since radicals and individual atoms possess a free valency, and can therefore form free-valency particles. Such reactions can proceed at a high rate. The start of a reaction between substances is preceded by a period during which active reaction centres in the form of charged particles accumulate in the medium, owing to the partial destruction of original molecules by other molecules which possess an energy higher than the energy of the atomic bonds in the original molecules. This is what is called the induction period.

The combustion of gaseous fuels. Combustion of gaseous fuels occurs by the laws of branched chain reactions which were discovered by Soviet Academician N. N. Semenov and C. N. Ilinshelwood. The conversion of the original substances to the final products passes through a sequence of reaction links which are connected in succession with one another and develop in the volume of a combustible mixture like the branches of a tree develop from its trunk. This results in the formation of the final reaction products and of an even greater number of active centres which further ensure the development of the reaction in the confining volume.

Let us consider the mechanism of branched chain reactions, taking as an example the combustion of hydrogen in air. By the stoichiometric equation

$$2H_2 + O_2 = 2H_2O$$

the rate of the reaction between molecules of the combustible substance

$$w_{\rm H_2O} = k_0 e^{-E/RT} C_{\rm H_2}^2 C_{\rm O_2} \qquad (4.9)$$

cannot be very large. Actually, however, combustion of hydrogen at temperatures above 500°C is an explosive chain reaction proceeding at a very high rate. Indeed, according to N. N. Semenov, the beginning of the

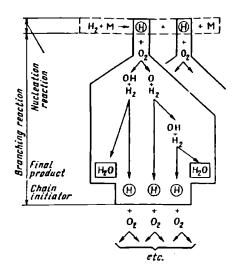


Fig. 4.3. Chain reaction cycle of hydrogen burning

— chain reaction exciter; — final product

active reaction is preceded by the formation of active centres:

$$H_2 + M^a \rightarrow 2H + M$$

 $H_2 + O_1^a \rightarrow 2OH$

where M^a and O^a are active molecules which possess high energy levels in the volume.

Atoms and radicals formed by this mechanism actively enter the reactions with the surrounding molecules, i.e. chains of successive reactions develop which result in the formation of the final reaction products and ever greater number of active centres.

Figure 4.3 schematically shows the first cycle of this reaction. As may be seen, each of the active hydrogen atoms H that has given rise to a chain reaction has produced three new active centres, owing to which the reaction progressively develops in the volume confining the gas mixture. As the reaction products accumulate and the concentrations of the starting substances become lower, chains are disrupted more often in the volume and at the walls of the reactor:

$$H + H \rightarrow H_2$$

 $OH + H \rightarrow H_2O$

The actual reaction rate is described by the equation:

$$w_{\rm H} = 10^{11} \sqrt{T} C_{\rm H} C_{\rm O_0} e^{-E^*/RT}$$
 (4.10)

The decisive factors for the reaction rate are the concentrations of hydrogen atoms (reaction centres) and oxygen molecules, with the activation energy E' of the reaction between them being substantially lower than E in equation (4.9). Similar laws of chain reactions govern the combustion of carbon monoxide CO, methane CH₄ and other combustible gases.

It follows from the foregoing that a short time, the induction period, precedes the beginning of an active reaction, during which a sufficiently large quantity of active centres (atoms and radicals) accumulates in the reaction volume. During this period, the reaction is almost unnoticeable and its thermal effect is negligible. After this period, the reaction rate increases due to the development of a large number of parallel reaction chains over the whole volume, until an equilibrium between the appearance and disappearance of active centres is established [36]. The reaction then attains its maximum rate and will proceed at this rate, provided that fresh portions of starting substances are regularly supplied to the combustion zone.

Combustion of a gaseous fuel in a mixture with air occurs at a very high rate (a ready methane-air mixture burns in a volume of 10 m³ in 0.1 s). For this reason, the intensity of combustion of natural gas in steam boiler furnaces is limited by the speed at which it mixes with air in the burner, i.e. by physical factors. The difficulties which arise when high flows of gas and air should be mixed thoroughly in a very short time in a burner are linked with the fact that the volume flow rates of the gas and air differ substantially, as approximately 10 m3 of air are needed for the combustion of 1 m³ of gas. For thorough intermixing, gas must be introduced into the air flow in the form of numerous fine jets and at a high rate. For the same pur-

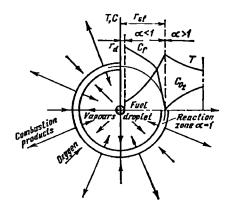


Fig. 4.4. Mechanism and combustion characteristics of a liquid fuel droplet

pose, the air flow is thoroughly turbulized by special swirling arrangements.

The combustion of liquid fuels. In the combustion of liquid fuels (petroleum, fuel oil), both the ignition and combustion temperatures (especially the latter) turn out to be higher than the boiling temperature of the individual fuel fractions. For this reason, liquid fuel first evaporates from the surface under the effect of the supplied heat, then its vapours are mixed with air, preheated to the ignition temperature and start burning. A stable flame forms at a certain distance from the surface of liquid fuel (0.5-1 mm or more).

Figure 4.4 schematically shows the combustion of a liquid fuel droplet in stagnant air. A vapour cloud forms around the droplet and diffuses into the environment, with the diffusion of oxygen of the air occurring in the opposite direction. As a result, the stoichiometric relationship between the combustible gases and oxygen is established at a certain distance r_{st} from the droplet, i.e. the burning fuel vapours form a spherical combustion front around it. The magnitude of rat is equal to 4-10 droplet radii, i.e. $r_{st} = 4-10r_d$, and depends heavily on the droplet size and the temperature in the combustion zone. In

the zone where $r < r_{st}$, fuel vapours prevail, but their concentration decreases inversely with the distance from the liquid surface. The zone with $r > r_{st}$ contains primarily combustion products mixed with the oxygen that has diffused into the combustion zone. The highest temperature is established in the reaction zone. Although at both sides of this zone the temperature decreases gradually, its decrease is more intensive in the inside direction, i.e. on approaching the droplet, since some heat is spent there for heating fuel vapours.

Thus, the burning rate of a liquid fuel droplet is determined by the rate of evaporation from its surface, the rate of chemical reaction in the combustion zone, and the rate of oxygen diffusion to this zone. As stated earlier, the reaction rate in a gaseous medium is very high and cannot limit the total rate of combustion. The quantity of oxygen diffused through the spherical surface is proportional to the square of sphere diameter, and therefore, a slight removal of the combustion zone from the surface of the droplet (under oxygen deficiency) noticeably increases the mass flow rate of supplied oxygen. Thus, the rate of combustion of the droplet is mainly determined by evaporation from its surface. The combustion rate of liquid fuels is increased by atomizing the fuel just before burning, which substantially increases the total surface of evaporation. Besides all this, as the size of the droplets decreases, the intensity of evaporation per unit area of their surface increases. Fine liquid fuel droplets suspended in an air flow move at low Reynolds numbers, Re $\ll 1$. In such cases, the heat flow through a spherical surface is determined solely by the conductivity \(\lambda\) through the boundary layer, which is much thicker than the droplet diameter. Under such conditions, the heat-transfer coefficient a is given by Sokolsky's formula:

$$Nu = \alpha d/\lambda = 2 \qquad (4.11)$$

whence

$$\alpha = 2\lambda/d = \lambda/r \qquad (4.12)$$

where Nu is the Nusselt number.

As follows from formula (4.12), the heat exchange between a droplet and the surrounding medium increases as the size of the droplet decreases, i.e. with a decrease in its mass. It turns out that the evaporation time of a droplet is proportional to the square of its initial diameter.

The combustion of solid fuel. When combined with air in a furnace, pulverized coal first passes through the stage of thermal preparation (Fig. 4.51), which consists in the evaporation of residual moisture and separation of volatiles. Fuel particles are heated up to a temperature at which volatiles are evolved intensively (400-600°C) in a few tenths of a second. The volatiles are then ignited, so that the temperature around a coke particle increases rapidly and its heating is accelerated (III'). The intensive burning of the volatiles (II) takes up 0.2-0.5 s. A high yield of volatiles (brown coal, younger coals, oil shales, peat), produces enough heat through combustion to ignite coke particles. When the yield of volatiles is low, the coke particles must be heated additionally from an external source (III"). The final stage is the combustion of coke particles at a tempera-

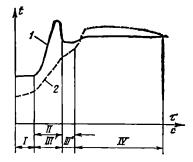


Fig. 4.5. Temperature conditions of burning of an individual solid fuel particle 1--temperature of gaseous medium around the particle; 2-particle temperature; I—thermal preparation zone; II—zone of burning of volatiles; III—heating of coke particle due to burning of its volatiles; III—beating of coke particle from an external source; IV—burning of coke particle

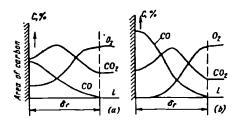


Fig. 4.6. Variations of concentration of gaseous substances at the surface of burning carbon

(a) burning at moderate temperatures; (b) burning at high temperatures; δ_f —thickness of boundary film

ture above 800-1 000°C (IV). This is a heterogeneous process whose rate is determined by the oxygen supply to the reacting surface. The burning of a coke particle proper takes up the greater portion ($^{1}/_{2}$ to $^{2}/_{3}$) of the total time of combustion which may constitute 1 to 2.5 s, depending on the kind of fuel and the initial size of particles.

The reacting mechanism between carbon and oxygen seems to be as follows. Oxygen is adsorbed from the gas volume on the surface of particles and reacts chemically with carbon to form complex carbon-oxygen compounds of the type C_xO_y which then dissociate with the formation of CO_2 and CO. The resulting reaction at temperatures near 1 200°C can be written as follows:

$$4C + 3O_2 = 2CO + 2CO_2 \qquad (4.13)$$

As has been established by experiment (L. Meyer, L. N. Khitrin), the ratio of the primary products, CO/CO₂, increases sharply with the increasing temperature of burning particles. For instance, the resulting equation at temperatures near 1 700°C can be written in the form:

$$3C + 2O_2 = 2CO + CO_2$$
 (4.14)

where the CO/CO₂ ratio is equal to two.

The primary reaction products are continuously removed from the surface of particles to the environment. In this process, carbon monoxide encounters the diffusing oxygen, which moves in the opposite direction, and reacts with it within the boundary gas film to be exidized to CO₂, with the result that the concentration of supplied exygen decreases sharply on approaching the surface of particles, while the concentration of CO₂ increases (Fig. 4.6a). At a high combustion temperature, carbon monoxide can consume all the exygen supplied, which, consequently, will not reach the solid surface of particles (Fig. 4.6b). Under such conditions, the endothermic reduction reaction will occur on the surface of particles, i.e. CO₂ will be partially reduced to CO.

Thus, heterogeneous combustion of a carbon particle from its surface can be represented as a process embracing four subsequent reactions (according to A. S. Predvoditelev), two of which are the main ones:

$$C + O_2 = CO_2 + q_1$$

 $2C + O_2 = 2CO + 2q_2$

the other two being secondary

$$2CO + O_2 = 2CO_2 + 2q_3$$

 $C + CO_3 = 2CO - q_4$

where q is the thermal effect of a reaction, MJ/mol.

The thermal effect of the first reaction $q_1 = q_2 + q_3$, while $q_4 = 0.57q_3$. The latter equation implies that even when the endothermic reaction takes place, the temperature of combustion is maintained at a rather high level due to a higher heat evolution in the volume.

As follows from an analysis of these reactions, the combustion of carbon from the surface takes place with partial gasification (formation of CO and its afterburning in the volume). This process accelerates the burning-off of coke particles.

4.3. Kinetic and Diffusion Regions of Combustion

As follows from an analysis of the mechanisms of solid and liquid fuel combustion, the total rate of combustion is determined not only by the rate of chemical reactions proper, but

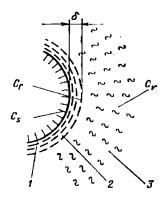


Fig. 4.7. Diagram of burning of carbon particle
t-particle surface; s-boundary film; s-turbutent flow region

also by the intensity of oxygen supply to the reaction zone by turbulent and molecular diffusion. This complex process can be analysed by using the following model of combustion (Fig. 4.7).

A burning particle is assumed to be surrounded by a boundary film within the boundaries of which afterburning of carbon monoxide takes place; mass transfer is assumed to be due to molecular diffusion only. Concentrations of substances and temperature vary within the film. Beyond that film, in the main flow around a particle, intensive turbulent mass transfer occurs. Therefore, the main flow is considered to be homogeneous, i.e. it has no gradients of temperatures and concentrations. It may be assumed in the final result that the oxygen concentration in the bulk of the main flow, C_{02}^b , is equal to the oxygen concentration at the film surface, C'_{O} .

The quantity of oxygen that passes by diffusion per unit surface area of particles is determined by Fick's law:

$$K_s^d = \alpha_d^r (C_{O_2}^h - C_{O_2}^s)$$
 (4.15)

where $\alpha_d = D/\delta$ is the mass-transfer coefficient which characterizes the rate of mass transfer of the substance through the layer of thickness δ , m/s, and D is the coefficient of molecular diffusion, m^2/s .

The highest supply of oxygen to the surface through the boundary layer will be observed at $C_0^* = 0$. Then

$$(K_a^d)_{\max} = \alpha_d C_{O_a}^b$$

To simplify the solution, let us assume that all the oxygen supplied by diffusion reacts on the particle surface. In that case the rate of the reaction per unit surface area in terms of oxygen consumption will be expressed by formula (4.5) as follows:

$$K_s^r = kC_{0s}^s \tag{4.16}$$

The maximum reaction rate will be at $C_{02}^{a} = C_{02}^{b}$

$$(K_s^r)_{\max} = kC_{O_q}^b$$

The pattern of variations of the maximum rates, $(K_s^d)_{\max}$ and $(K_s^t)_{\max}$, is shown in Fig. 4.8. Noting that the oxygen consumption in the surface reaction must be equal to the oxygen supply through the boundary film, i.e. $K_s^d = K_s^r = K_s$, equations (4.15) and (4.16) can be solved together to give:

$$K_{\bullet} = \frac{\alpha_d k}{\alpha_d + k} C_{0\bullet}^b = k_r C_{0\bullet}^b$$
 (4.17)

The structure of this equation is similar to that of a common kinetic equation with the reaction rate constant k being replaced by the resolved coefficient k, with due consideration given to the reactivity of the fuel and the conditions of oxidant supply to the reacting surface.

Temperature has only a weak effect on mass transfer by diffusion (a_d~ $\sim T^{0.5}$), but determines to an appreciable extent the rate of reaction on the surface $(k \infty e^{-E/RT})$. The nature of variation of the maximum rates of oxygen diffusion to the surface and oxygen absorption by that surface (Fig. 4.8) suggests two principal regions of combustion of coke particles. At relatively low temperatures (less than 1 000°C for coke particles), the surface reaction proceeds rather slowly and oxygen consumption is only a small fraction of the quantity of oxygon that can be supplied to the sur-

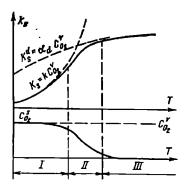


Fig. 4.8. Regions of constant-size particle burning and variations of oxygen concentration at its surface

I-III—respectively kinetic, transition and diffusion region of burning

face, i.e. $k \ll \alpha_d$. In that case $k_r \approx k$ and $C_{02}^b \approx C_{02}^c$. Then

$$K_{\bullet} = kC_{\mathrm{O}_{2}}^{\bullet} \tag{4.18}$$

i.e. the total rate of reaction is limited by the kinetics of the chemical reacting on the surface. This temperature region of reactions is called the kinetic combustion zone. In this zone, the combustion process is independent of the conditions of oxygen supply.

At high temperatures (above 1400°C), the rate constant of the reaction on the surface increases rapidly and, at a certain moment, exceeds the maximum rate of oxygen supply to the surface, which varies only slightly with temperature. Then, $k \gg \alpha_d$ and the total reaction rate is determined by the rate of oxygen supply:

$$K_a = \alpha_d C_{O_2}^b \tag{4.19}$$

In this zone, the reaction rate varies slowly, notwithstanding the increasing temperature. The oxygen supplied to the surface by diffusion reacts instantaneously and its concentration at the surface becomes virtually equal to zero. The reaction might proceed at an appreciably higher rate, but is retarded by an insufficient supply of oxygen. This temperature region is called the diffusion combustion zone. With oxygen deficiency at the surface, the reaction of the reduction of

CO₂ to CO takes place at the incandescent coke surface, while the diffusing oxygen is completely consumed in the boundary film by gaseous CO where this is oxidized to carbon dioxide CO₂.

In this zone, the rate of combustien increases with the increasing rate of gas flow around particles and with the decreasing size of solid particles.

At intermediate temperatures (1 000°-1 400°C), the rate of reaction at the surface becomes commensurable with the rate of oxygen supply, and the total rate of the reaction is determined by both processes. This is called the transition zone of combustion. Its position in the temperature scale depends on the size of coke particles. With smaller particles, the transition zone appears at a higher temperature.

The ignition of any fuel begins at relatively low temperatures under conditions of ample oxygen supply, that is, essentially in the kinetic zone. As the temperature rises, the oxygen consumption in the reaction zone rapidly increases and the process passes through the transition zone into the diffusion zone. The end of combustion is usually well in the diffusion zone where the residual concentrations of the oxidant turn out to be rather low, substantially retarding the burning-off of the remaining fuel [29].

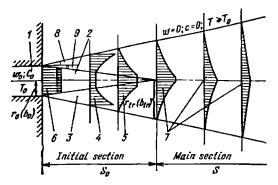
4.4. The Ignition of Fuel-air Mixture. Combustion Front

In power engineering, chamber furnaces are the main type of furnace used for high-capacity steam boilers. The fuel-air mixture from a burner enters the furnace in the form of straight or swirled jets whose development in the furnace space determines the conditions of ignition and combustion intensity.

Let us first consider the mechanism of development of a straight jet which flows into the furnace space filled with furnace gases (Fig. 4.9). The jet flows from the burner embrasure with the initial parameters as follows: velo-

Fig. 4.9. Structure of free turbulent jet development

1—end portion of nozzle (burner); g—jet core; 3—boundary layer; 4—temperature distribution in the jet; 5—distribution of concentrations of combustible matter in the jet; 6—velocity diagram at burner outlet; 7—velocity diagrams in the main region; 5—jet spread angle; 9—internal angle



city w_0 , temperature T_0 , and fuel concentration C_0 . At the interface between the jet and environment, transverse components of pulsation velocity cause the penetration of the jet mass into the surrounding volume. In the mixing zone, which is called the boundary layer of a jet, the interaction of the masses obeys the law of conservation of motion:

$$M_0 w_0 + M_1 w_1 = (M_0 + M_1) w_m$$
 (4.20)

where M_0 and M_1 are the interacting masses of the jet and environment and w_0 and w_1 are their velocities; under the conditions considered, $w_1 = 0$.

In view of this, the average velocity of translatory motion of the mixture, w_m , may be found as

$$w_m = \frac{M_0 w_0}{M_0 + M_1} \tag{4.21}$$

On approaching the external boundary of the jet, the proportion of the mass entrained in motion, M_1 , increases, and the velocity of the mixture w_m decreases.

Owing to the turbulent mass transfer, the boundary layer gradually widens to the inside as well as to the outside, so that its potential core, which retains the initial parameters w_0 , T_0 , C_0 (undisturbed portion of the jet), gradually diminishes. The external angle of jet expansion is equal to 12-14°, while the internal angle of expansion of the boundary layer is around 6°. The section in which the potential core disappears is called the

transient section and the distance from it to the burner end is called the initial section of jet, S₀. Downstream of the transient section, the boundary layer extends across the whole jet, so that the parameters at the jet axis begin to vary with distance from the transient section (jet velocity decreases and its temperature increases). The length of the initial section of the jet is found as:

$$S_0 = 0.67 \frac{r_0}{a}$$
 (4.22)

where r_0 is the initial radius of a round jet or 1/2 of the equivalent diameter of a jet with a rectangular cross section, and a is the experimental coefficient of the jet structure, which accounts for its initial turbulence and the non-uniformity of the initial velocity field.

In a turbulent flow with Reynolds number Re > 2×10^4 , the coefficient a = 0.07-0.08, whence the length of initial section is $S_o = (4.2-5) d_{eq}$, where d_{eq} is the equivalent diameter of a jet. In the transient section, the half-width, depending on the external angle of jet expansion, will be

$$r_{tr} = r_0 + S_0 \tan \alpha_{ex} \qquad (4.23)$$

thus, it exceeds roughly 3 times the initial jet radius.

The supplied fuel is heated to the ignition temperature by two heat sources: by the radiant heat flux from the flame core and, more important, by convective heating due to the intermixing of the jet mass with hot furnace gases.

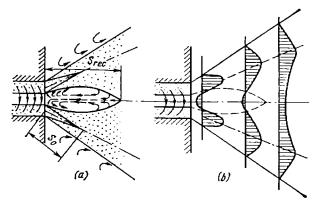


Fig. 4.10. Structure of swirled annular jet at burner outlet (a) general view; (b) axial velocity distribution; S_0 —length of initial region; S_{rec} —length of gas recirculation region

In view of this, burners should be designed so as to intensify as much as possible the entrainment of hot furnace gases into the fresh jet in the initial section, thus accelerating fuel ignition. An example of such a burner is the vortex burner with an annular swirled jet (Fig. 4.10). Swirling of the jet is determined by the vorticity parameter n=2-5, which is found as the ratio

$$n = \frac{4M}{KD} \tag{4.24}$$

where M is the angular momentum relative to the jet axis, K is the projection of the angular momentum vector onto the jet axis, and D is the characteristic size of the jet; for round (continuous or coaxial) jets, the characteristic size is their equivalent diameter d_{eq} .

The vorticity parameter can be approximately found from the formula:

$$n \cong \frac{4w_1}{w_a} \tag{4.25}$$

where w_t is the maximum tangential component of the flow velocity at the burner exit, and w_a is the axial velocity component [3, 57].

An increase of the *n* parameter results in greater turbulization of the jet, more intensive entrainment of the surrounding gases into it, and a wider expansion angle.

A turbulized jet has a zone of reduced pressure in its core near the axis into which high-temperature gases may tend to move, meaning that recirculation of gases to the jet root may take place. The length of the recirculation zone depends on the vorticity parameter: $S_{rec} = 1.4nr_0$. Thus, a swirled annular jet differs from a straightflow jet mainly in higher turbulence and in the appearance of an internal zone for entrainment of hot gases into the jet, whereas a straight jet has only an external zone of intermixing. For these reasons, a turbulized jet heats up more quickly.

For the ignition of a hot mixture, two conditions are essential: the temperature of the combustible mixture must be not less than the ignition temperature and the concentration of the combustible substance must exceed the lower inflammability limit.

At the external boundary of a jet where the temperature t_m is the highest, the concentration of the combustible substance is insufficient for ignition. On the other hand, in the core of the jet, the temperature is too low for ignition. Therefore, combustion can start only in a rather narrow jet band near the periphery where both conditions are fulfilled.

The combustion front is characterized by an elevated turbulization due

to a sharp increase in the volume of gases. The combustion front will be stable if fresh portions of fuel and air are continuously supplied to it. Combustion always takes place at a certain distance from the burner end, since the jet just near the burner does not possess the required temperature. The combustion front is established in a place where the translatory velocity of the flow turns out to be equal to the velocity of propagation of a turbulent flame.

4.5. The Burn-off Intensity of Fuel

The initial stage of fuel burning takes place at a high concentration of the combustible substance and oxidant and an elevated turbulence of the flow which is formed by the burner. The zone of the furnace within which intensive combustion of fuel takes place to a burn-off degree $\psi_t = 0.85 - 0.9$ is called the flame core zone. It can be characterized by a high temperature and substantial thermal radiation onto the surrounding water walls (Fig. 4.11). The size of the flame core zone may constitute $\frac{1}{3}$ to $\frac{1}{5}$ of the furnace space volume. The remaining portion of the furnace volume is what is called the zone of fuel afterburning and gas cooling. It is characterized by a low residual concentration of fuel and oxidant and a weak gas flow turbulence, because of which fuel afterburning (burn-off) takes place deep

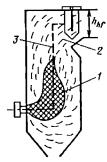


Fig. 4.11. Zones of fuel burning in furnace space

t—flame core zone; 2—zone of fuel afterburning and gas cooling; 3- resolved flame length; $h_{h\,t}$ height of horizontal gas duct

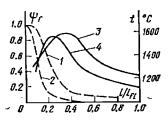


Fig. 4.12. Fuel burn-off and gas temperature variations along the flame height 1—anthracite burn-off characteristic; 2—fuel-oil burn-off characteristic; 3—gas temperature on anthracite combustion; 4—gas temperature on fuel-oil combustion

in the diffusion region and proceeds

slowly.

The degree of fuel burn-off in a furnace space is usually related to the resolved flame length lfl, which is understood as the horizontal distance from the burner end to the furnace axis plus the vertical distance from the burner level to the level of the horizontal gas duct plus the horizontal distance to the furnace outlet (see Fig. 4.11). Figure 4.12 shows curves of fuel burnoff and temperatures in a furnace which depend on the relative flame length in combustion of solid fuels and fuel oil obtained by the results of tests of high-capacity steam boilers. As follows from the figure, the burning of a solid fuel is essentially completed at the relative flame length $l/l_{fl} =$ = 0.35 and that of fuel oil at l/l_{fl} = = 0.25. In the latter case, the flame core disappears in the initial horizontal portion of the resolved flame length.

Pulverized coal is characterized by polydisperse particle size distribution. Its burning begins from the finest fractions which are heated up to the ignition point in a few hundredths of a second. As the fine particles burn, they accelerate the heating of the larger particles; but these begin burning when the major portion of oxygen has already been consumed. For this reason combustion of larger particles takes place mainly in the diffusion region and they are mainly responsible for the loss of fuel as unburned carbon

in waste gases.



COMBUSTION PRODUCTS

5.1. The Composition of Combustion Products

A knowledge of the composition of a fuel and of the reactions by which its combustible components are oxidized makes it possible to calculate the volume of combustion products obtained. In boiler plant operation, combustion products are analysed continuously or periodically in order to control the completeness of fuel combustion and the density of gases in the waste gas path.

In thermal calculations of boiler plants, the composition and temperature of flue gases are used to determine the enthalpy downstream of each heating surface. It is also essential to know the volume of gases for certain aerodynamic calculations.

Combustion reactions at high temperatures proceed at a high rate. Therefore, the composition of the final combustion products is close to the equilibrium value. In zones where oxygen has been deficient, products of incomplete oxidation of the original combustible components may remain in fuel gases. The composition of combustion products formed on the combustion of 1 kg of solid or liquid fuel or 1 m³ of gaseous fuel can be written in the following form:

Fuel (1 kg) + air
$$\rightarrow$$

$$\frac{V_{\text{CO}_2} + V_{\text{SO}_2} + V_{\text{H}_2\text{O}}}{1} + \underbrace{\frac{V_{\text{N}_2} + V_{\text{O}_2} + V_{wv}}{2}}_{2} + \underbrace{\frac{V_{\text{CO}} + V_{\text{H}_2} + V_{\text{CH}_4}}{3}}_{3}$$
(5.1)

where $V_{\rm CO_1}$, $V_{\rm SO_2}$, etc. are the volu-

mes of individual gas components in combustion products, m³/kg or m³/m³*.

Combustion products of a fuel can be conveniently divided into three groups. Group 1 in formula (5.1) includes the products of complete combustion of fuel elements, i.e. dry triatomic gases usually denoted as V_{RO}.

$$V_{\rm RO_2} = V_{\rm CO_2} + V_{\rm SO_2}$$
 (5.2)

and water vapours $V_{\rm H,O}$. In the composition of dry triatomic gases, $V_{\rm CO}$. is always much greater than $V_{
m SO, ,}$ since fuels commonly have a low sulphur content. Group 2 contains the volumes of nitrogen and oxygen which represent the residual dry air left upon fuel combustion and water vapours. In this group, $V_{
m N_{
m s}}$ is much larger than V_{O_1} , since oxygen has been consumed in a large amount for oxidation. The volume of water vapours includes the evaporated moisture of the fuel and the moisture content of the air proper. Group 3 denotes the products of incomplete combustion of fuel elements; in this group, $V_{
m CO}$ > $>V_{\rm H},>V_{\rm CH}$. The ratio of volumes $V_{\rm CO}:V_{\rm H}$, is 3:1 on the average. The presence of V_{CH} in the products of incomplete combustion is indicative of strong deviations from the specified conditions of combustion.

Let us analyse complete fuel combustion, assuming the combustion products have $V_{\rm CO}=0$, $V_{\rm H_1}=0$, $V_{\rm CH_4}=0$ and no residual oxygen, i.e. $V_{\rm O_r}=0$. The quantity of air required for the complete combustion of 1 kg (1 m³) of fuel to utilize the oxygen of the air

In further discussion, only unit volumes and enthalpies per kg fuel will be used.

without residue is called the theoretically required air volume V^0 . In that case, according to formula (5.1), the theoretical volume of combustion products is

$$V_g^0 = V_{\text{CO}_3} + V_{\text{BO}_2} + V_{\text{H}_2\text{O}}^0 + V_{\text{N}_2}^0$$
 (5.3)

We must distinguish between the theoretical volume of dry gases:

$$V_{dg}^{0} = V_{CO_{3}} + V_{SO_{2}} + V_{N_{2}}^{0} = V_{RO_{3}} + V_{N_{3}}^{0}$$
 (5.4)

and the total volume of gases:

$$V_g^0 - V_{dg}^0 + V_{H_2O}^0 \tag{5.5}$$

In this case, the volume $V_{11,0}^{0}$ includes all the water-vapour components in combustion products which have been considered earlier, while the volume V_{N}^{0} , is formed mainly from the nitrogen of the air, with a slight addition of the nitrogen of the fuel which evolves from the latter when heated together with other volatile components.

The volume of dry triatomic gases $V_{\rm RO_2}$ in formulae (5.2) and (5.4) is the same and does not depend on whether the theoretical air volume V^0 or a greater quantity V_a is supplied for burning*, while the volumes of other components of combustion products will change. In this case, the volumes corresponding to the theoretical conditions of combustion are marked by the superscript '0'.

When the combustion of fuel is complete and there is any excess quantity of air in the combustion zone, i.e. with $V_a > V^o$, there forms the same theoretical volume of gases V_g^o . However, the actual gain in the volume of combustion products is determined solely by the excess quantity of air $\Delta V_a = V_a - V^o$ and by the water vapours contained in it, $\Delta V_{^a}|_{I_aO}$.

Under real conditions, fuel cannot be burned completely at the theoretically sufficient air volume V^0 since the short time the gases are present in

the furnace (1-2 s) is not sufficient for the thorough intermixing of fuel and air in the large volume of the furnace. For this reason, the actual volume of air that can ensure complete fuel combustion is always somewhat greater than the theoretical value. The ratio of these volumes is called the excess air coefficient (ratio) in combustion products:

$$\alpha = V_a/V^0 \tag{5.6}$$

The required excess air ratio of a furnace, a_f , depends on the kind of fuel, method of combustion, and furnace design. High-reactive solid fuels with a high yield of volatiles are ignited more easily and burn off more quickly, and therefore, require a lower excess air ratio than those having a low yield of volatiles. In gas-air mixtures, fuel can be intermixed effectively with air, for which reason fuel oils and gaseous fuels require a still lower air excess. The same grade of fuel may require different excess air ratios when burned in various furnaces (say, in a straight-jet and vortex furnace) which differ in effectiveness of fuel-air intermixing.

The rated excess air ratio α_f can be found according to the Norms of thermal calculation of steam boilers by considering all the factors involved [20]. For the various kinds of fuel, it can usually be taken within the following limits:

A decrease in the excess air ratio may save energy for driving blowers and increase the boiler efficiency, but its reduction below the rated value α , may result in a sharp rise in heat loss with unburned carbon and a lower boiler efficiency.

In steam boilers with supercharged furnaces, the excess air ratio at the exit from the furnace, α_f , is equal to its value in the burner, α_b , and remains unchanged along the whole gas path, since all gas ducts in this case

^{*} The concentrations of CO₂ and SO₂ in atmospheric air are insignificant and disregarded in calculations.

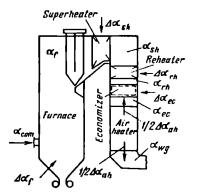


Fig. 5.1. Inleakage of cold air into boiler

operate under an excess pressure and are made gas-tight (except for the regenerative air heater).

In steam boilers operating at a negative pressure (which is formed by induced-draft fans), cold air from the surroundings can leak into the gas path through untight joints between various elements (cracks and joints in the lining, places where the tubes pass through the lining, etc.). For this reason, the volume of combustion products increases, thus increasing the excess air ratio, while the temperature of gases decreases (Fig. 5.1). Inleakage can be found as a fraction of the theoretically required air volume:

$$\Delta \alpha_i = \frac{\Delta V_i}{V^0} \tag{5.7}$$

where ΔV_i is the volume of sucked-in air within a particular *i*-th heating surface of the boiler.

In such a case, the air inleakage downstream of the *i*-th heating surface after the furnace will be found as:

$$\alpha_i = \alpha_i + \sum_{i=1}^{i} \Delta \alpha_i \qquad (5.8)$$

Air inleakage may also take place in the furnace proper $(\Delta \alpha_f)$. Noting this, the excess air ratio in the combustion zone will be:

$$\alpha_{com} = \alpha_f - \Delta \alpha_f \qquad (5.9)$$

The volume of waste gases downstream of the last heating surface of the

boiler can be found using the formula:

$$V_{wg} = V_c^0 + (\alpha_{wg} - 1) V_c^0$$
 (5.10)

As may be seen, it consists of the volume of products of complete combustion V_c^* and the total excess air ΔV_{ex} whose amount increases on the path of motion of waste gases in flue ducts. In turn, the total amount of excess air can be divided into two components:

$$\Delta V_{ex} = (\beta_{com} - 1) V^0 + \Sigma \Delta \alpha_l V^0$$
(5.11)

where $\beta_{com} = \alpha_{com}$ is the excess air ratio at the burner exit.

The first term in formula (5.11) is the excess air that is deliberately added to ensure complete combustion of fuel, and the second term represents harmful air inleakage.

The volume of air for complete combustion of solid and liquid fuels is determined by writing down the reactions involved in the combustion of fuel elements [20, 37]. In the combustion of carbon and sulphur, the volumes of CO_2 and SO_3 which form V_{CO_2} and V_{SO_2} , turn out to be equal to the volume of oxygen consumed in combustion: $V_{RO_2} = V_{O_3}$ and, since the volume of nitrogen remains practically unchanged, $V_{dg}^0 = V^0$.

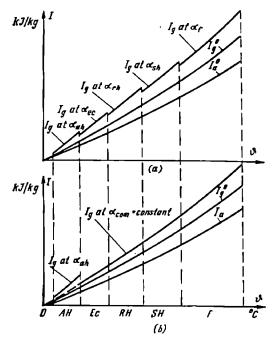
When the hydrogen of fuel is burning, the volume of water vapours formed is twice the volume of the oxygen consumed. Thus, the theoretical volume of gases, V_g^0 , of a fuel containing hydrogen is always greater than the theoretically required air volume V^0 , even when 'dry' fuel devoid of adventitious moisture is burned.

For gaseous fuels, the consumption of oxygen for combustion is calculated for each of the gas components separately, noting the percent age of their concentration in the gaseous fuel.

The total volume of air and flue gases is found as the product of the volume calculated for 1 kg (m³) of the fuel by the total consumption of fuel. In addition, the method of resolved thermal characteristics can be used for

Fig. 5.2. I, 0-diagram of air and combustion products

(a) in operation at reduced pressure; (b) with supercharge; AH—air heater; Ee—economizer; RH—reheater; SH—superbeater; F—furnace



the purpose, in which case the calculated values are related to the unit of heat liberated by the fuel (1 000 kJ or 1 MJ). The development and application of this method is largely due to the work of Prof. S. Ya. Kornitsky, M. B. Ravich, and Ya. L. Pecker [45]. The method is based on the Welter-Berehier rule which states that the quantity of heat liberated on combustion of dry fuel is directly proportional to the quantity of consumed oxygen, i.e. $Q^c = a' V_{0_s}$. Thus, with a fuel of the known heating value and moisture content, it is possible to determine the theoretical volume of air for combustion and the volume of combustion products.

The enthalpy of combustion products is found for the volume of gases obtained on combustion of 1 kg or 1 m³ of fuel. It is denoted as I, kJ/kg or kJ/m³. Since the specific heats of various gases are different, enthalpies are calculated separately for each component of flue gases. For instance, the enthalpy of the theoretical volume of combustion products at a tem-

perature &, °C, is:

$$I_g^0 = (V_{RO_2}c_{RO_3} + V_{N_2}^0c_{N_2} + V_{H_2O}^0c_{H_2O} + a_{ash} \frac{A^w}{100} c_{ash}) \vartheta$$
 (5.12)

where c_{RO_s} , c_{K_s} , c_{H_sO} are the mean specific volume heats of flue gas components in the temperature range 0-0, $kJ/(m^3 \cdot K)$, c_{ash} is the specific heat of ash particles, $kJ/(kg \cdot K)$, and a_{ash} is the relatively quantity of ash particles carried off by flue gases.

The last term in equation (5.12), which characterizes the enthalpy of ash particles carried off by flue gases, is considered only for the combustion of high-ash fuels.

The enthalpy of gases at an excess of air $\alpha > 1$ can be found by the equation:

$$I_g = I_g^0 + \Delta I_a = I_g^0 + (\alpha - 1) I_a^0$$
(5.13)

where I_a^0 is the enthalpy of the theoretical volume of air:

$$I_a^0 = V^0 c_a \vartheta \tag{5.14}$$

where c_a is the volume specific heat capacity of air, $kJ/(m^3 \cdot K)$.

Calculated enthalpies of gases at various temperatures can be plotted as an I, 0-diagram, which is convenient for practical use (Fig. 5.2).

5.2. Determination of Excess Air Ratio for an Operating Boiler

The excess air ratio in the gas path of a boiler plant should be continuously controlled in order to provide the optimal conditions for fuel combustion and to minimize air inleakage. Two such methods of control are currently in use at power stations. The most popular method is to directly measure the residual oxygen in the flue gas flow by means of oxygen meters. This method is based on using the magnetic properties of oxygen melecules, a property that is not possessed by other components in a multicomponent gas mixture.

The percentage of oxygen can be recalculated into the excess air ratio in the following manner. Neglecting the slight increase in the volume of combustion products due to the liberation of nitrogen from the fuel, it may be taken that the volume of dry gases is $V_{dg} = V_a$. The residual oxygen in combustion products can be expressed as a fraction (per cent) of the volume of dry gases:

$$O_2 = \frac{21 (\alpha - 1) V^0}{V_{dg}}$$
 (5.15)

As has been stated earlier, $V_{dg} = \alpha V^0$, and therefore:

$$O_2 = \frac{21(\alpha-1)}{\alpha}$$
 (5.16)

and the excess air ratio is:

$$\alpha = \frac{21}{21 - O_3} \tag{5.17}$$

where O_2 is the concentration of oxygen in combustion products, %, as measured by an oxygen meter.

This formula holds true whon fuel combustion is complete and air humidity can be neglected. The necessary corrections should be introduced if the flue gases contain products of incomplete combustion (CO, H₂).

Another popular method for determining the excess air in combustion products is based on finding the percentage of dry triatomic gases $RO_2 = CO_2 + SO_2$ in combustion products:

$$RO_2 = \frac{V_{RO_2}}{V_{de}} 100 (5.18)$$

With complete fuel combustion in the stoichiometric ratio ($\alpha = 1$) and provided that the content of hydrogen and oxygen in fuel correspond to the equality $H^w = O^w/8$, all the hydrogen in the fuel is considered to be oxidized by the oxygen of the fuel, and therefore

$$RO_2^0 = \frac{V_{RO_2}}{V_{dg}'} 100 = 21 \%$$

In all solid and liquid fuels, however, $H^{\omega} > O^{\omega}/8$. The residual hydrogen $\Delta H^{\omega} = H^{\omega} - O^{\omega}/8$ will then be oxidized by atmospheric oxygen and form water vapours. The remaining volume of nitrogen, ΔV_{N_s} , enters the composition of dry gases, and the maximum concentration of dry triatomic gases will be less than 21%:

$$RO_{a}^{max} = \frac{V_{RO_{2}}}{V'_{dg} + \Delta V_{N_{2}}}$$

$$\times 100 = \frac{V_{RO_{2}}}{V_{dg}^{0}} 100 \qquad (5.19)$$

and will depend on the difference between H^w and O^w/8.

The difference in the concentrations of hydrogen and oxygen can be expressed in terms of the fuel coefficient β . For solid and liquid fuels, β is found by the formula:

$$\beta = 2.37 \frac{H^{\omega} - O^{\omega/8}}{C^{\omega} + 0.375S_{o+p}^{\omega}}$$
 (5.20)

where $S_{\nu}^{w}_{+p}$ is the sum of organic and pyrite sulphur in the working mass of fuel.

Using the fuel coefficient β, RO ax can be written as follows:

$$RO_1^{max} = \frac{21}{1+\beta} \qquad (5.21)$$

Gaseous fuels contain no oxygen, but their hydrogen content is substantially higher, and therefore, RO^{max} turns out to be much lower than that of solid and liquid fuels.

The RO₂^{max} values of various fuels are within the following limits:

Solid fuels	-					
		•				16-17%
Natural gas						11-13%

With $\alpha > 1$, the volume of dry gases will be:

$$V_{dg} = V_{dg}^0 + \Delta V_{dg}$$

and the volume of triatomic gases is

$$\mathrm{RO_2} = \frac{V_{\mathrm{RO_2}}}{V_{dg}^0 + \Delta V_a} \, 100$$

i.e. smaller than RO_1^{max} . Since V_{RO_2} remains unchanged, we have:

$$\frac{RO_{a}^{max}}{RO_{a}} = \frac{V_{dg}^{0} + \Delta V_{a}}{V_{dg}^{0}} = \frac{V_{a}}{V^{0}} = \alpha \quad (5.22)$$

The accuracy of the excess air ratio α determined by this method depends on how accurately we know the RO_2^{max} of the fuel (since the composition of fuel delivered to a power station is not always perfectly constant) and how accurately the flue gases have been sampled and analysed for RO_2 .

Certain difficulties in the application of this method for determining α arise in the combustion of fuels containing carbonates in their mineral portion as they dissociate thermally with the evolution of an additional, often variable, volume of CO_2 .

5.3. Toxic Substances in Waste Gases and Measures of Environmental Control

Thermal power stations consume around 40% of the total fuel produced in the world and thus can substantially influence the environment by ejecting heat and certain harmful pollutants into the atmosphere and water basins. This necessitates systematically protecting the environment from the harmful effluents of indu-

strial plants for the sake of both the present generation and those of the future.

Certain components in the gaseous effluents from power stations are safe for man; these include water vapours, carbon dioxide, oxygen and nitrogen. Other ingredients are harmful to varying degrees.

Thermal power stations and large boiler houses pollute the atmosphere by ejecting nitrogen oxides $(NO_x =$ = NO + NO₂), sulphur oxides (SO₂ + + SO₃), solid ash particles, and vanadium pontoxido V₂O₅. Moreover, if combustion is incomplete, flue gases may contain carbon monoxide CO, hydrocarbons, such as CH4 or C2H4, benzpyrene C₂₀H₁₂, and soot. The length of time the harmful gaseous substances may exist in the environment ranges from a few hours to a few days. Many are gradually destroyed or transformed into harmless substances under the offect of solar radiation and atmospheric oxygen. Air-borne solid particles (soot, vanadium pentoxide, benzpyrone) may settle and accumulate on the earth's surface and be involved into the circulation of the lower atmospheric layers.

By their effect on the human organism, harmful substances are divided into a number of classes. Vanadium pentoxide and benzpyrene relate to extremely dangerous substances. The former appears in minor quantities on the combustion of fuel oils. Benzpyrene may appear in waste gases on the combustion of any fuel with oxygen deficiency and is also formed on the slow decomposition of soot. Nitrogen dioxide NO2 and sulphuric anhydride belong to the class of highly dangerous substances. Nitrogen oxides NO_x mainly form in the high-temperature zones of the flame. The yield of NO2 constitutes around 10% of the total yield of NOx. Sulphur of fuel mainly burns off to form sulphurous anhydride SO2 which, like NO, is placed in the third class of harmful substances by their effect on man (moderatoly harmful substances). Sulphuric anhydride SO₃ may form in furnaces from SO₂ at the final stage of flame burning when there is an excess of atomic oxygen or by catalysis on the high-temperature deposits of steam superheaters. Its yield is 2-5% the yield of SO₂. In the zone of low-temperature heating surfaces, SO₃ is transformed into sulphuric acid vapours and is partially spent in the process of low-temperature corrosion (see Ch. 16).

The risk from the effect of a particular substance on a living organism is determined as the ratio of the actual concentration of the substance C, mg/m³ to the highest permissible concentration (HPC), mg/m³, in air at the breathing level:

$$k_i = C_i / \text{HPC}_i \tag{5.23}$$

This ratio is called the toxic rate of an *i*-th substance. It should always be less than unity.

If air contains a number of harmful substances with a similar biological effect on the living organism, its toxicity is enhanced. Therefore, the presence of such substances in air is inadmissible even in concentrations near the HPC level. In such a case, the toxic hazard is estimated by adding together the toxic rates of the individual substances. For waste gases, the additivity principle is applied to NO₂, NO, SO₂, SO₃, and CO. Noting this, the additivity formula of toxic effluents is as follows:

$$\frac{c_{\text{NO_2}}}{\text{HPC}_{\text{NO_2}}} + \frac{c_{\text{NO}}}{\text{HPC}_{\text{NO}}} + \frac{c_{\text{SO_2}}}{\text{HPC}_{\text{SO_2}}} + \frac{c_{\text{SO_3}}}{\text{HPC}_{\text{SO_3}}} + \frac{c_{\text{CO}}}{\text{HPC}_{\text{CO}}} \leqslant 1 \quad (5.24)$$

In many cases, henzpyrene is included into this formula. For solid (particulate) effluents, the additivity formula is:

$$\frac{C_a}{\text{HPC}_a} + \frac{C_s}{\text{HPC}_a} + \frac{C_{\text{V}_2\text{O}_5}}{\text{HPC}_{\text{V}_2\text{O}_5}} \leqslant 1 \quad (5.25)$$

where the subscripts 'a' and 's' stand for 'ash' and 'soot'.

The highest permissible concentrations of selected harmful substances are given in Table 5.1.

Table 5.1. Highest Permissible Concentrations of Selected Harmful Substances in Air, mg/m³

		-				
	<u> </u>	In inhabited area				
Substance	Formula	highest single concen- tration	average dally concen- tration			
Nitrogen dioxide Nitrogen oxide Sulphuric anhyd- ride Sulphurous anhyd- ride Carbon monoxide Benzpyrene	NO ₂ NO SO ₃ SO ₂ CO C ₂₀ H ₁₂	0.085 0.25 0.30 0.50 3.0	0.085 0.25 0.10 0.05 1.0 1×10-6			
Vanadium pento- xide Soot (grit) Dust (ash particles)	V ₂ O ₅	0.15 0.50	0.002 0.05 0.15			

Power stations account for roughly 3/4 of the total sulphur oxides ejected into the atmosphere. In view of the increasing industrial utilization of high-sulphur fuel oils and solid fuels, reducing the total discharge of sulphur oxides in the atmosphere is an urgent concern. Since the volume concentration of sulphur oxides in waste gases is rather low (around 0.2%) and the volumes of waste gases are enormous, purification of gases from SO2 is difficult to accomplish. It seems more reasonable to purify petroleum from sulphur at petroleum processing plants or to subject fuel oil and highsulphur solid fuels before combustion to a deep thermal processing which can produce gaseous fuel and valuable liquid products. In such processes, sulphur compounds can be extracted and collected much more easily. Such methods of fuel processing are being used on an ever increasing scale.

The extraction of NO_x from waste gases involves even larger engineering difficulties than that of SO_2 . This is due to a lower concentration of NO_x in gases (which is only 1/2 to 1/5 of that of SO_2) and the lower chemical activity of the NO which is present in

the gases in a much larger concentration than NO_2 . Around 40% of the total quantity of nitrogen oxides ejected into the atmosphere is from power stations [47].

In contrast to gaseous SO₂ whose presence in waste gases is uncontrollable and depends only on the concentration of sulphur in the fuel, the concentration of nitrogen oxides in waste gases depends heavily on combustion organization in the furnace. Nitrogen oxides are mainly produced from the atmospheric oxygen and nitrogen in the high-temperature combustion zone (at temperatures above 1 600°C) and by the oxidation of the nitrogen of fuel (this reaction can occur at temperatures below 1 600°C). Thus, the main means of reducing the formation of NOx in furnaces are to reduce the temperature in the combustion zone ('low-temperature' combustion) and limit the quantity of excess air. It should be noted, however, that with improper operation of the furnace or burners, these measures may result in the appearance of other equally harmful substances (such as soot or benzpyrene).

Around 20% of the total contamination of the atmosphere with solid dust particles falls on ash discharge by power stations. The discharge of solid fractions by power stations may incre-

ase in the future due to the increasing power of stations and the increasing use of solid fuels. In this respect, the problem of how deeply to purify flue gases from ash particles is growing in magnitude. Fly ash formed by the combustion of solid fuels contains a number of harmful components, such es fluoride and arsenic compounds and the oxides SiO₂ and CaO. Ash collection at thermal power stations is most often carried out in electrostatic precipitators. For many fuels with an elevated content of ash, the degree of ash collection must be no less than 99%.

For the present, until efficient industrial plants capable of removing harmful impurities from fuel or from waste gases are developed, one method of decreasing the concentration of toxic substances in the lower atmosphere is to build tall stacks up to 320 m high which can spread flue gases to large distances. With higher stacks, the concentration of harmful impurities in the atmosphere at the earth's surface decreases in inverse proportion to the square of the stacks' height ratio. Tall stacks are not, however, a radically effective means of protecting the environment from industrial effluent; they simply diminish the local concentrations of harmful substances in the lower atmosphere.



EFFICIENCY OF FUEL HEAT UTILIZATION

6.1. The Heat Balance and Efficiency of the Steam Boiler

The efficient utilization of fuel in steam boilers is primarily determined by the following three factors:

- (1) complete combustion of the fuel in the hoiler furnace:
- (2) deep cooling of the combustion products during their passage through the heating surfaces; and
- (3) minimization of heat losses to the environment.

The extent to which the heat of the fuel is transferred in the boiler to the working fluid is determined by the boiler's gross efficiency. This is defined as the quantity of heat absorbed by the working fluid Q_1 , kJ/kg of solid or liquid fuel or kJ/kg of gaseous fuel, related to the available heat of the working mass of fuel, Q_{ab}^{ab} , kJ/kg:

$$\eta_b = \frac{Q_1}{Q_{nn}^w} 100 \tag{6.1}$$

The available heat of burned fuel is determined by the general formula:

$$Q_{iv}^{\omega} = Q_i^{\omega} + Q_{ph} + Q_{aex} + Q_{sb} - Q_{cd}$$
 (6.2)

where Q_i^w is the lower heating value of the working mass of fuel, kJ/kg, Q_{nh} is the physical heat of solid or liquid fuel before burning, kJ/kg, Q_{aex} is the quantity of heat absorbed by the supplied air during preheating outside the boiler (most often in external air heaters) kJ/kg, Q_{ab} is the heat of steam used in steam burners for fuel oil atomization, kJ/kg, and Q_{cd} is the heat from the dissociation of carbonates in the mineral mass of fuel, kJ/kg (the last term is applicable only to oil shales).

The physical heat of fuel before burning depends on its temperature

$$Q_{ph} = c_t t_t \tag{6.3}$$

where c_f is the specific heat of fuel, kJ/(kg K), and t_f is its temperature, °C.

The average temperature of solid fuels may range from 0 to 20°C, but may be even sub-zero in winter time. In the combustion of fuel oil, the term Q_{ph} must always be taken into account since, for atomization, fuel oil is preheated in the burner up to 100-130°C.

The quantity of heat absorbed by the air for combustion during preheating outside the boiler is found from the formula:

$$Q_{a\sigma x} = \beta_a V^0 c_a (t'_a - t_{ca}) \quad (6.4)$$

where βa is the relative air quantity passing through a heater, ca is the

specific heat of air, t_{ca} is the temperature of cold air, and t'_a is its temperature at the entry to the boiler air heater (or behind the external air heater).

The heat introduced by steam for

fuel oil atomization is:

$$Q_{sb} = d_{sb} \left(i_{sb} - i_{s}' \right) \tag{6.5}$$

where d_{ab} is the unit flow rate of steam for atomization, kg/kg fuel (usually $d_{ab} = 0.05$ -0.1 kg/kg), i_{ab} is its enthalpy, kJ/kg, and i_{s} is the enthalpy of steam carried off by flue gases at a temperature ϑ_{wg} , kJ/kg.

The quantity of heat that is spent on the dissociation of carbonates in oil shales is proportional to the quantity of carbon dioxide that evolves on combustion, CO_{cd}^{cd} , so that the formula for determining Q_{cd} is:

$$Q_{cd} = 40.5 \text{CO}_2^{cd}$$
 (6.6)

To sum this up, the heat available on the combustion of various kinds of fuel can be determined by the following formulae:

— for anthracites, coals and brown coals of a low humidity and low sulphur content:

$$Q_{av}^{w}=Q_{l}^{w};$$

— for moist brown coals and for coals and fuel oil with a high sulphur content:

$$Q_{ac}^w = Q_l^w + Q_{ph} + Q_{aex};$$

— for fuel oil atomized by steam in burners:

$$Q_{av}^w = Q_i^w + Q_{ph} + Q_{aex} + Q_{ab};$$

— for natural gas:

$$Q_{av}^{\omega}=Q_{l}^{d};$$

- for oil shales:

$$Q_{av}^{w} = Q_{l}^{w} - Q_{cd}.$$

The heat absorbed by the working fluid (water and steam) in the boiler per kg (or m³) of fuel can be found from the formula

$$\begin{aligned} Q_{1} &= \frac{D_{hh}}{B} \left(i_{ss} - i_{fw} \right) + \frac{D_{rh}}{B} \left(i_{rh}^{*} - i_{rh}^{*} \right) \\ &+ \frac{D_{bw}}{B} \left(i^{*} - i_{fw} \right) \quad (6.7) \end{aligned}$$

where D_{sh} and D_{rh} are the flow rate of fresh superheated steam and steam secondary superheating (reheated steam), kg/s, D_{bw} is the flow rate of blow-off water from the boiler drum to maintain the required salt conditions in the circulation circuit, kg/s, D_{bw} must be considered if it is more than 2% of D_{sh} ; l_{ss} , i_{fw} , and i'are respectively, the enthalpy of superheated steam, feed water and water at the saturation line at the pressure in the drum, kJ/kg, i_{rh} and i_{rh} are the enthalpies of secondary superheated steam at the inlet to and outlet from the reheat superheater, kJ/kg, and B is the fuel consumption, kg/s or m³/s.

The heat utilized in a boiler can also be expressed in terms of heat absorption by the individual heating surfaces:

$$Q_i = Q_f + Q_{sh}^{con} + Q_{rh} + Q_{ee}$$
 (6.8) where Q_f is the heat absorption by

the working fluid in the furnace water walls, kJ/kg, Q_{ch}^{con} is the heat absorption in the convective surfaces of superheater, kJ/kg, Q_{rh} is the heat absorption in the secondary superheater, kJ/kg, and Q_{ec} is the heat ab-

sorption in the economizer, kJ/kg.

During boiler operation, some of the available heat is inevitably lost and constitutes heat losses. The distribution of the heat supplied to a boiler as useful heat and lost heat is the basis for compiling the heat balance of a steam boiler. The heat balance equation corresponds to steady-state thermal conditions of boiler operation. It is usually written for 1 kg or 1 m³ of burned fuel:

$$Q_{av}^{w} = Q_1 + \underbrace{Q_2 + Q_3 + Q_4 + Q_5 + Q_6}_{\text{heat loases}} (6.9)$$

Dividing both sides of equation (6.9) by Q_{av}^{w} and expressing the result as a percentage, we get:

$$100 = q_1 + q_2 + q_3 + q_4 + q_6 + q_6$$
 (6.10)

The typical items of the boiler heat balance and their values are given in Table 6.1.

Table 6.1. Ileat Losses in Steam Boilers

Heat loss item	Relative loss, % of Quo	q, %
With waste gases Through incomplete combustion	q2 q3	4-7 0-0.5
With unburned car-	q_4	0.5-5
bon By cooling through	q_{5}	0.2-1
the lining As physical heat of	q _e	0-3
removed slag Sum of heat losses	Σq_l	6-12

The direct calculation of the boiler efficiency by formula (6.1) may turn out to be insufficiently accurate and involve difficulties in accurate measurement of certain parameters, e.g. mass flow rates of steam and fuel, heating value of fuel, and some components of the available heat.

The gross efficiency of a steam boiler can be found by the inverse balance method if we know the sum of its heat losses:

$$\eta_b = 100 - (q_2 + q_3 + q_4 + q_5 + q_6)$$
(6.11)

The inverse balance method, i.e. determining the boiler efficiency by the sum of heat losses, gives a more accurate value than the direct balance method, since the sum of heat losses constitutes roughly 1/10 of Q_{av}^{av} and all its items can be reliably measured. This is the sole method available for estimating the thermal efficiency of new steam boilers at the design stage.

If we know the heat losses and, consequently, the gross boiler efficiency, we can use formulae (6.1) and (6.7) to find the fuel consumption by the boiler, kg/s:

$$\frac{B}{D_{sh}(i_{ss}-i_{fw})+D_{rh}(i_{rh}^{r}-i_{rh}^{r})+D_{bw}(i_{r}^{r}-i_{fw})}{Q_{nv}^{w}\eta_{b}}$$
(6.12)

The fuel consumption thus obtained is the rated value for calculating the

pulverization system of the boiler. In solid fuel-fired boilers, more often than not, fuel is not burned completely, i.e. there is a heat loss with unburned carbon q_4 . To find the actual volumes of combustion products and the air flow rate for combustion, the concept of rated fuel consumption is introduced:

$$B_r = B (1 - 0.01q_4) (6.13)$$

The gross efficiency characterizes the perfection of operation of a steam boiler proper. Its normal operation, however, depends in addition on a large number of auxiliary machines and mechanisms, which consume a portion of the energy produced by the power plant (station). The use of energy for this purpose is called the auxiliary power consumption, Naux, of a boiler plant. The auxiliary power consumption includes the power consumed by forced- and induced-draft fans, feed pumps, pulverizing equipment, blowing devices, and a large number of electric motors in the automatic remote control system. For boilers operating in monobloc units with turbines, the auxiliary power consumption can be found as a percentage of the total power production by the following formula:

$$\Delta \eta_{aux} = \frac{N_{aux}}{BQ_{av}^{\omega} \eta_{st}} 100 \quad (6.14)$$

where B is the fuel consumption by the boiler, kg/s, and η_{st} is the efficiency of power production at the station.

In high-capacity boilers, $\Delta \eta_{aux}$ is equal to 4-5%. By subtracting the auxiliary power consumption $\Delta \eta_{aux}$ from the gross efficiency of a boiler η_b , we obtain what is called the net boiler efficiency, which characterizes the efficiency of the boiler plant operation in terms of the electric energy delivered to consumers:

$$\eta_b^{net} = \eta_b - \Delta \eta_{aux} \qquad (6.15)$$

6.2. Analysis of Heat Losses

Heat loss with waste gases. According to Table 6.1, heat loss through waste gases is the largest item in the heat balance. Its relative magnitude, per cent, can be found by the formula:

$$q_2 = \frac{Q_2}{Q_{av}^{w}} 100 \tag{6.16}$$

where Q_2 is the absolute heat loss, $kJ/k\sigma$:

$$Q_2 = (I_{wg} - \alpha_{wg}I_{ea}^0) (1 - 0.01q_4)$$
(6.17)

Here, I_{wg} is the enthalpy of waste gases, kJ/kg, and I_{ca}^{o} is the enthalpy of the theoretically required volume of cold air, kJ/kg. The first multiplier in the equation gives the gain in enthalpy of waste gases relative to that of the air supplied to the boiler (both organized and unorganized). The second multiplier is a correction for incomplete combustion of fuel. As follows from formula (6.17), the heat loss through waste gases Q_2 is determined, first of all, by the enthalpy of waste gases I_{wg} , i.e. it depends on their temperature ϑ_{wg} and their volume which is characterized by the excess air ratio α_{wg} .

By reducing the temperature of the waste gases by 15-20 deg C, it is possible to decrease q_2 and thus raise the boiler efficiency by roughly 1%. This can be achieved by arranging an additional convective heating surface ΔH_c whose dimensions will be the larger, the greater the reduction in the waste gas temperature (Fig. 6.1). It should also

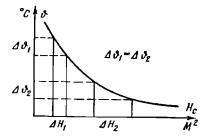


Fig. 6.1. Variation of the additional heating surface area in various temperature regions

be noted that ϑ_{wg} depends substantially on the temperature of the feed water t_{fw} at the inlet to the economizer and the temperature of air at the inlet to the air heater, t'_a . This can be expressed by the following relationship $\{6\}$:

$$\vartheta_{\omega_{\mathcal{L}}} = \frac{\omega - 1}{\omega} \left(t_{fw} + \Delta t_2 \right) + \frac{t_a' + \Delta t_1}{\omega} \tag{6.18}$$

where $\omega = \Sigma (Vc)_g/(V_ac_a)$ is the ratio of the water equivalents of the gases and air in the lower stage of the air heater, Δt_1 is the temperature gradient (between gas and air) at the 'hot end' of the air heater, and Δt_2 is the temperature gradient (between gas and water) at the 'cold end' of the economizer.

The recommended minimum values of temperature gradients [20] are $\Delta t_1 = 30 \deg C$ and $\Delta t_2 = 40 \deg C$; but somewhat higher values are usually taken in the boiler design, since the values indicated require excessively large surfaces of air heater and economizer. The ratio of water equivalents depends on the quality of fuel and the excess air ratio in the air heater zone.

For a particular kind of fuel and the selected values of Δt_1 , Δt_2 and excess air ratio, formula (6.18) takes the form:

$$\vartheta_{\omega g} = A + Bt_{f\omega} \qquad (6.19)$$

As follows from the above, a complex engineering-economical approach is essential for selecting the optimal temperature of waste gases ϑ_{wg}^{opt} . A change of ϑ_{wg} will substantially influence the size of the heating surfaces of the economizer, air heater and regenerative heaters. This circumstance will change the hydraulic resistance of the gas and water paths and the energy consumption by feed water pumps and blowers. A probable change in the height of the stack to better disperse the waste gases should also be considered.

Thus, if measures are taken to lower the temperature of the waste gases

and feed water with the temperature of hot air remaining the same, the additional expenditures involved can be found as follows:

$$Z = \Delta S_{ah} + \Delta S_{ec} - \Delta S_{wh} + \Delta S_{bl} - \Delta S_{fu} + \Delta S_{al}$$
 (6.20)

where ΔS_{ah} and ΔS_{ec} are the costs of the additional heating surfaces of an air heater and an economizer, ΔS_{bl} is the extra cost of electric energy for driving the blowers, associated with the increased resistance of the air-gas path, ΔS_{sl} is the cost for constructing a taller stack, ΔS_{wh} is the saving in feed water heaters, and ΔS_{fu} is the fuel savings.

The optimal temperature ϑ_{up}^{opt} is found by minimizing the calculated expenditures, which is done by solving the equation:

$$\partial Z/\partial \vartheta_{\omega g} = 0 \tag{6.21}$$

Typical curves of the optimal temperature of waste gases as functions of the decisive economic factors are shown in Fig. 6.2. The optimal temperature of waste gases depends substantially on the cost and quality of fuel, and, first and foremost, on its moisture content. When the cost of fuel is high, the savings in consumed fuel can justify the extension of heating surfaces and thus allows a lower temperature of waste gases (Fig. 6.2a). A high moisture content in the fuel increases the volume of combustion products and their specific heat, since water vapours possess the highest specific heat in these gases. To cool by the same number of degrees $\Delta \vartheta_{wg}$, waste gases with a high humidity must give up a greater quantity of heat, which requires a further increase in the heating surfaces compared with dry fuel. For the same or lower priced moist fuel, extending the heating surfaces is not economically feasible, which means that the optimal temperature of the waste gases will increase along with the moisture content of the fuel (Fig. 6.2b).

An increase in the initial parameters of steam (pressure and temperature)

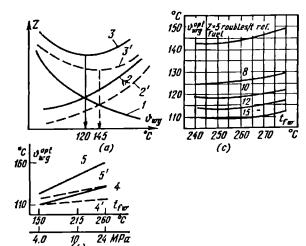


Fig. 6.2. Determination of optimal temperature of waste gases (a) dependence on the cost of heating surfaces and fuel; I—expenditures on heating surfaces; 2 and 2'—expenditures on more expensive and less expensive fuel; 3 and 3'—total calculated expenditures with more expensive and less expensive fuel; (b) dependence on the feed water temperature and the moisture content of fuel; 4 and 4'—limits for dry fuels with $W^2 < 0.7$; 5 and 3'—ditto, for moist fuels with $W^2 = 1.5$; (c) correlation between the optimal temperature of waste gases, the temperature of feed water and the cost of fuel for a supercritical pressure monoblec unit

can significantly raise the efficiency of heat regeneration, which justifies a certain increase in the feed water temperature. On the other hand, increase in the initial pressure of steam increases the expenditures on boiler equipment as well as energy consumption by the feed water pumps. As shown earlier, an increase in the temperature of feed water, t_{fw} , results in a higher waste gas temperature 0wg, and therefore, in higher cost and lower economic efficiency of the boiler plant. For this reason, thermodynamic possibilities for increasing the efficiency of boiler plants are always restricted. As follows from the calculated relationships (Fig. 6.2c), the effect of t_{fw} on \mathfrak{d}_{wg}^{opt} is more pronounced with the less expensive fuels.

In boiler plant design, practical values of $\vartheta_{\omega g}$ are usually taken in the range of 120-160°C.

As regards semi-peak load boiler plants, which are put in operation only during periods of elevated electric loads, the problem of minimizing boiler cost is more important than that of decreasing the waste gas temperature. For this reason, boilers of this type are characterized by lower steam parameters and higher temperature of waste gases ($\vartheta_{wg} = 160\text{-}200^{\circ}\text{C}$).

When selecting the optimal waste gas temperature, one should also consider the probable corrosion of low-temperature heating surfaces — especially the surface of the air heater (see Sec. 16.3). To forestall this, boilers to be fired on high-sulphur fuels (Somore than 2%) are designed for a higher temperature of waste gases (up to 140-160°C) and with additional preheating (to 60-80°C) of the air supplied to the air heater. Certain measures are also taken to minimize corrosion in the air heater (see Sec. 19.4).

Heat loss by incomplete combustion. The products of combustion may contain gaseous combustible components, such as CO, H_2 or CH_4 . Their afterburning beyond the boiler furnace is practically impossible since the temperature of gases and the concentrations of the combustible components and oxygen are too low. The heat that might be produced by afterburning these components constitutes what is known as the heat loss by incomplete combustion Q_3 , kJ/kg, or q_3 , %. It can be calculated by the formula [59]:

$$q_3 = (126.4 \text{CO} + 108 \text{H}_2 + 358.2 \text{CH}_4)$$

$$\times \frac{V_{dg} (100 - q_4)}{O_{gg}^{w_g}} \qquad (6.22)$$

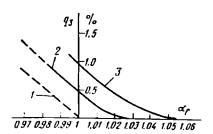


Fig. 6.3. Heat losses due to incomplete combustion

where CO, H_2 and CH₄ are the volume concentrations of the products of incomplete combustion in dry gases, %, V_{dg} is the volume of dry gases, m³/kg fuel, $(100-q_4)$ is the degree of combustion of the solid portion of fuel, %, the numbers at the symbols of gases are their respective heating values, kJ/m³, divided by 100.

On burning gaseous and liquid fuels, the heat loss by incomplete combustion $q_a = 0.0.5\%$. With solid fuels, this item is, as a rule, rather small and neglected. In boiler practice, this item of heat losses is mainly determined by the concentration of CO and, to a lesser extent, H₂ in combustion products. The appearance of CH₄ in combustion products is indicative of an improperly organized combustion. Analysis for incomplete combustion should always be done for all the components of combustion products since, as follows from formula (6.22), even a slight quantity of CH, may have a noticeable effect on q_3 .

Heat loss from incomplete combustion substantially depends on the excess air ratio and boiler load (Fig. 6.3). Thoroughly intermixing the fuel and oxygen (curve 1) ensures that heat loss by incomplete combustion may take place only at $\alpha < 1$. Under real conditions (curve 2), the appearance of this loss at $\alpha = 1$ and full boiler load is indicative of an improper intermixing of fuel and air. Incomplete combustion cannot take place at what is called the critical excess air ratio α_{cr} , which is usually equal to 1.02-1.03 and thus characterizes the

aerodynamic perfection of the burner arrangement. With the boiler operating at a reduced load (curve 3), the exit rate of the fuel and air from the burner decreases (thus decreasing the energy for intermixing the fuel and air flows) and the temperature level in the combustion zone diminishes somewhat, which results in an increased heat loss from incomplete combustion.

Heat loss with unburned carbon. In the combustion of solid fuels (peat. coals and oil shales), the heat loss with unburned carbon is essentially the loss with unburned coke particles carried off from the combustion zone by furnace gases; during the short time they are present in the high-temperature zone of the flame, these particles evolve volatiles, but remain partially unburned. In the combustion of fuel oil and natural gas, the unburned carbon loss may be in the form of solid particles (which remain upon the evaporation of fuel oil droplets) or soot particles which can form in high-temperature combustion zones at a deficiency of oxygen ($\alpha < 0.6$).

Under normal operating conditions, the unburned carbon heat loss of solid fuels q_4 may range from 0.5% to 5%, with the lower values relating to fuels with a high yield of volatiles (peat and brown coal), and the higher ones to those of a low reacting ability (low yield of volatiles), such as anthracite. For coals, $q_4 = 0.5-2\%$. The heat loss q_4 from the combustion of natural gas and fuel oil is not high (usually less than 0.1%) and is considered along with q_3 , i.e. as the sum $q_3 + q_4$.

In chamber-furnace combustion of solid fuels, the heat loss with unburned carbon Q_4 , kJ/kg, is divided into the carry-over loss Q_4^{co} and the loss with slag Q_4^{cd} , the former being predominant.

The carry-over loss is determined by fine coke particles which are carried off (over) by the gas flow from the furnace, channeled through all the boiler flue ducts and, together with ash particles, are collected in electrostatic precipitators or other types of

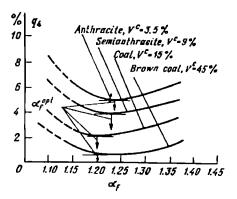


Fig. 6.4. Heat losses with unburned carbon in dry-bottom furnaces

dust collectors and discharged to ash dumps. The loss with slag is due to the fact that some unburned fuel particles in the combustion zone are fused with molten slag and drop onto the furnace bottom.

When the optimal fineness of fuel pulverization is obtained and under normal operating conditions of the boiler, the loss q_4 depends on excess air and varies widely with changes in the yield of the volatiles (Fig. 6.4). When the excess air ratio is below the optimal level, q_4 may rise owing to an improper intermixing of fuel and air at the exit from the burner or to the appearance of an oxygen deficiency zone though the temperature level developed by the burner is quite high. With $\alpha > \alpha_{opt}$, the temperature in the combustion zone decreases, resulting in a lower rate of oxidation reactions (see Ch. 4). On the other hand, the volume of combustion products increases, thus diminishing the time the particles are present in the hightemperature zone. These two factors are responsible for an increase in the heat loss with unburned carbon.

An elevated value of q_4 in low-reactive fuels (anthracite, semianthracite) is due to the delayed ignition of coke particles and prolonged burning in the diffusion region. These kinds of fuel are, therefore, rather sensitive to variations in operating conditions. As an example, Fig. 6.5 shows the effect

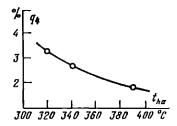


Fig. 6.5. Effect of hot air temperature on heat loss with unburned carbon

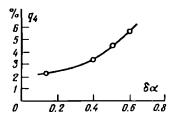


Fig. 6.6. Variations of unburned carbon heat losses in slagging-bottom furnaces as a function of the degree of non-uniformity of air distribution between burners at $t_{ha} = 400^{\circ}\text{C}$

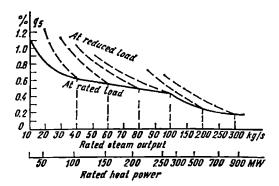
of variations of hot air temperature and Fig. 6.6 shows the effect of a non-uniform distribution of air between the burners on the unburned carbon heat loss q_4 ; in the latter figure, $\delta \alpha$ is the relative deviation of excess air ratio from the rated value.

Heat loss by cooling. Since the temperature of the lining and casing of a boiler and of its other elements (such as drum, headers, pipelines, etc.) is higher than that in the surrounding air, they give up heat to the environment. In general, the heat loss by cooling, Q_5 , kJ/kg, can be expressed by the formula:

$$Q_5 = \frac{F_{bw}}{B_r} \left(\alpha_c + \alpha_r \right) \left(t_{bw} - t_{sur} \right) \quad (6.23)$$

where F_{bw} is the exposed surface area of the walls and the high-temperature elements of the boiler, m^2 , α_c and α_r are the coefficients of heat transfer by convection and radiation, $kW/(m^2 K)$, and t_{bw} and t_{sur} are respectively, the average temperature of the

Fig. 6.7. Heat losses due to external cooling



boiler surfaces and the surrounding air, °C.

The heat loss by cooling is the higher, the higher the external temperature of the lining and heat insulation of the boiler. The external surfaces of boilers and boiler elements should be properly heat-insulated so that t_{biv} is not higher than 55°C. In rough calculations, the heat flux from the boiler surfaces to the surroundings is taken at an average level $q_{II} = 200-300$ W/m2. For boilers with a different power rating, the absolute heat loss to the surroundings, Q_5 , is determined by the ratio F_{bw}/B_r . With increasing boiler power, the absolute heat loss Q_5 , kJ/kg, and the relative loss q_5 , %, become lower, because the total heat release and the volume of combustion products increase more quickly than the area of exposed boiler surfaces (Fig. 6.7).

As is seen from Fig. 6.7, the heat loss item q_5 for high-capacity boilers is not high. It is commonly assumed that these losses are proportional to the heat absorbed by each heating surface of the boiler and are accounted for by what is called the heat retention coefficient.

tion coefficient:

$$\varphi = 1 - \frac{q_3}{\eta_b + q_3} \qquad (6.24)$$

In this formula, the ratio $q_5/(\eta_b + q_5)$ characterizes the relative heat loss to the outside.

In boilers of a lower capacity, the absolute heat loss through enclosing walls remains virtually unchanged, and therefore, the relative heat loss increases:

$$q_5 = q_5^r \frac{D_r}{D} \tag{6.25}$$

where the subscript (superscript) 'r' relates to rated values. Heat losses q_5 at $D < D_r$ are shown by dotted lines in Fig. 6.7.

Heat loss with physical heat of slag. The slag removed from the bottom of a boiler furnace has a rather high temperature, and therefore, possesses a certain quantity of heat which is transferred to cooling water in the slag bath and is lost irreversibly. This constitutes the heat loss with the physical heat of slag, Q_6 . The relative loss, q_6 , can be found by the formula (%):

$$q_6 = \frac{a_{sl} (ct)_{sl} A^w}{O_{cr}^w} \tag{6.26}$$

where $a_{sl} = 1 - a_c$ is the fraction of the total ash that is removed as slag from the furnace, $(ct)_{sl}$ is the enthalpy of slag, kJ/kg, and Λ^w is the ash content in the working mass of fuel.

The heat loss q_6 can be determined graphically, as in the graph in Fig. 6.8, where $A^r = A^w/Q_1^w$ is the resolved ash content of fuel. The heat loss q_6 depends substantially on the method of slag removal from the boiler furnace (see Ch. 7). In dry-bottom furnaces, $a_{el} = 0.05$ -0.1 and the temperature of the slag is 600-700°C; in that case and with a relatively low ash content in the fuel, q_6 is insigni-

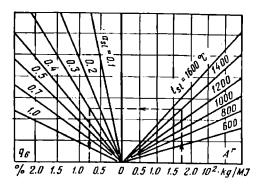


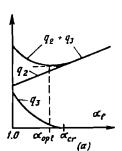
Fig. 6.8. Heat losses with physical heat of removed slags

ficant (Fig. 6.8). This heat loss is relevant only to high-ash fuels with a resolved ash content A^r greater than 2.5. In slagging-bottom furnaces, the temperature of flowing slag is on the average $t_{al} = 1\,400\text{-}1\,600^\circ\text{C}$. The fraction of ash removed through the furnace bottom also increases up to $a_{al} = 0.15\text{-}0.3$ in single-shaft furnaces and up to as much as 0.5-0.7 in those with primary cyclone furnaces. In such cases, the heat loss q_6 should be given due consideration.

As follows from the analysis of heat losses, some heat loss items depend significantly and in different ways on the excess air ratio (q_2, q_3, q_4) . It is therefore essential to establish the optimal excess air ratio for a furnace which will minimize the total heat

losses. In this case, the heat loss q_2 is related to the excess air ratio in the furnace, α_f , assuming that inleakages in the flue ducts are constant: $\Sigma \Delta \alpha_i = \text{constant}$. Figure 6.9a shows a typical curve of heat losses $q_2 + q_3 = f(\alpha_f)$ in the combustion of natural gas and fuel oil when the heat loss q_4 can be neglected. As we can see, the optimal excess air ratio in the furnace turns out to be close to α_{cr} and is always slightly lower than α_{cr} since the curve $q_2 = f(\alpha)$ is steeper in the zone near α_{cr} .

In the combustion of solid fuels (Fig. 6.9b), the heat losses q_2 and q_4 have a vital effect on the optimal excess air ratio. In that case, α_{opt} is somewhat smaller than the value α corresponding to the minimum of q_4 .



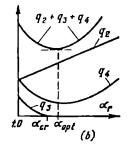


Fig. 6.9. Determination of the optimal excess air ratio in the furnace by the minimum of heat losses
(a) compution of natural gas and fuel oil:

(a) combustion of natural gas and fuel oil;
(b) combustion of solid fuel

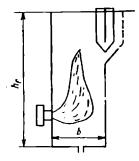


PULVERIZED COAL-FIRED FURNACES

7.1. Principal Characteristics of Chamber Furnaces

In the furnace the chemical energy of fuel is converted into the heat of combustion products as the pulverized fuel (coal dust) is burned in suspension in the furnace space. Combustion products give up part of their heat by radiation to water walls. Thus they leave the furnace at a safe temperature which will not cause clinkering of the subsequent convective heating surfaces.

A furnace can be characterized geometrically by its linear dimensions: front width a, depth b, and hoight h_f (Fig. 7.1) which are calculated according to the rated fuel consumption and the thermal and physico-chemical characteristics of the fuel to be used.



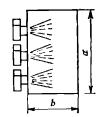


Fig. 7.1. Main dimensions of furnace

The product $f_f = ab$, m^2 , is the cross-sectional area of a furnace through which hot furnace gases pass at a rather high speed (7-12 m/s). An enormous quantity of heat is released in a furnace at the level of the burners, and the temperature of gases in it increases substantially.

The principal thermal characteristic of steam boiler furnaces is what is called the heat power of furnace, kW:

$$Q_I = BQ_I^w \tag{7.1}$$

i.e. the quantity of heat released in the furnace at the rated consumption B, kg/s, of fuel with the heating value Q_I^{w} , kJ/kg. Relating the total heat release in the combustion zone of a furnace to its cross-sectional area, we obtain another important characteristic called the heat release rate per unit furnace area:

$$q_t = Q_t/f_t \qquad (7.2a)$$

The highest allowable value of q_f is assigned depending on the kind of fuel and the arrangement and type of burners used. This value may range from 3 500 kW/m² for clinkering coals and brown coals up to 6 400 kW/m². An increase of q_f results in an increase in the flow rates and temperatures of gases in the furnace as well as heat absorption by the furnace water walls.

If the burners are arranged in a number of tiers, the furnace can also he characterized by the heat release rate per burner tier:

$$q_t^t = Q_t/f_t^s \tag{7.2b}$$

where Q_t is the heat released by all the burners of a tier, kW.

For various fuels, q may range from 1 200 kW/m² to 2 300 kW/m².

An increase of q_j and q_j^t above their limiting values can result in intensive clinkering of the water walls, especially in the burner zone, and in a dangerous rise in temperature of the tubes' metal. The rated value q^r is taken to be slightly lower than the highest allowable limit. When the heat power of a furnace is known [from formula (7.1)], the required cross-sectional area of the furnace chamber can be determined as:

$$f_f = Q_f/q_f^r \tag{7.3}$$

The depth b of a furnace is chosen in the range of 6 to 10.5 m so as to properly arrange the burners (see Sec. 7.2) on the furnace walls and ensure free development of the flame in the furnace space such that the high-temperature flame tongues do not touch the water walls. The depth of the furnace is increased up to 8-10.5 m when using more powerful burners with larger ports arranged in several tiers (two or three) on the furnace walls.

The front width of the furnace a is equal to 9.5-31 m depending on the kind of fuel and the thermal power (steaming capacity) of the boiler and can be calculated from the known values of f_f and b. As the boiler capacity increases, so does a although not in direct proportion. This characterizes an increase of the heat release rate and gas flow rate in higher capacity boilers. The front width of the furnace can be found from the formula:

$$a = 0.67 \sqrt{D} \tag{7.4}$$

where D is the steaming capacity of the boiler, 1/h.

The height h, of the furnace is taken to be in the range of 15 to 65 m to ensure virtually complete combustion of the fuel along the flame length within the furnace and allow space for arranging the water walls on the furnace walls to cool the combustion products to the specified temperature. The furnace height required to ensure complete fuel combustion can be found from the formula:

$$h_f^{\rm com} = \overline{w}_g \tau_{pr} \tag{7.5}$$

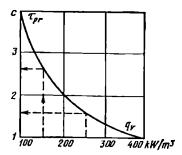


Fig. 7.2. Correlation between the heat release rate per unit furnace volume and the time of presence of gases in furnace

where \overline{w}_g is the average velocity of gases in the furnace cross section, m/s, and τ_{pr} is the time a unit volume of gas is present in the furnace, s.

It is essential that the time τ_{pr} be much larger than τ_{com} , the time for complete combustion of the largest fractions of fuel (see Sec. 4.2).

The thermal conditions in the volume of a furnace can be characterized by what is called the *allowable* heat release rate, or energy release rate, per unit volume kW/m³:

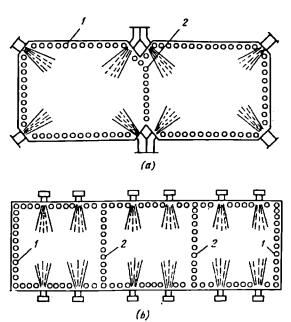
$$q_V = Q_t/V_t = BQ_t^w/V_t$$
 (7.6)

where V_f is the furnace volume, m^3 . The values of allowable heat release rate per unit volume are standardized. They may range from 120 kW/m³ for coal-fired dry-bottom furnaces 210 kW/mⁿ for slagging-bottom furnaces. The magnitude of q_V determines the average time the gases are present in the furnace space. With an increase in q_V , the time of gas presence in the furnace decreases (Fig. 7.2). The highest allowable value of q_v corresponds to the equality $\tau_{pr} =$ au_{com} , which also determines the lowest allowable volume of a furnace Vinin_

As indicated earlier, water walls in a furnace should cool the combustion products to the specified temperature at their exit from the furnace space, 07, which is achieved by properly selecting the dimensions of water walls,

Fig. 7.3. Diagrams of furnaces with (a) one and (b) two platens:

J-water walls; 2-platens



and consequently, the volume of the furnace.

To this end, it is reasonable to compare the minimal volume of a furnace V_t^{\min} as found from the condition of complete combustion of fuel with its value V_i^c as determined by the conditions of cooling of the gases to the specified temperature of. As a rule, $V_{l}^{c} > V_{l}^{\min}$ for all solid fuels, and therefore, the furnace height should be determined by the conditions of gas cooling to the specified temperature 07. In many cases, the required furnace height as found by calculations substantially exceeds the minimum value obtained from its volume V_t^{\min} (especially with coals that have an elevated value of external ballast), resulting in an excessively high boiler mass which in turn substantially increases expenses. On the other hand, as the steaming capacity of boiler increases, the furnace volume increases relatively more substantially than the surface area of the water walls. Therefore, the unit surface area of cooling surfaces per m³ of furnace volume becomes lower,

which explains why an increase in boiler capacity involves an increase in the temperature 07.

The cooling surface area of a boiler can be increased without changing the furnace dimensions by arranging curtain walls, or platens (Fig. 7.3) additional tube walls mounted in the furnace space and dividing this into two or more sections. In high-capacity boilers with a large width of furnace, the latter can be divided by platens into nearly rectangular sections, which facilitates fuel combustion and produces a more uniform field of gas temperatures and heat release rates on the water walls. In contrast to conventional water walls on the sides of a furnace, a curtain wall is heated intensively from both sides and is characterized by a higher heat release rate, which necessitates more careful cooling of the tube metal.

The arrangement of curtain walls can noticeably reduce the height and volume of the furnace, bringing both closer to the minimal allowable values. Comparable characteristics, with and without a platen, for a 300-MW

monobloc boiler plant operating on anthracite at the rated furnace temporature $\vartheta_i^* = 1.150^{\circ}\text{C}$ are given in Table 7.1.

Table 7.1. Effect of Platen on Boiler Furnace Characteristics

Characteristic	Without platen	With platen
Minimal furnace vo-	5 190	5 190
Furnace volume designed by cooling conditions,	8 478	6016
V'/V'min ratio	1.63	1.16
Designed furnace height, h_f , m	48.0	36.2

To obtain the best ratio between the radiant and convective heat transfer in boiler surfaces, it is advisable to maintain the temperature of furnace gases at the furnace outlet near 1 250°C. For most solid fuels, however, this is not possible since the gas temperature at the outlet from the furnace space to convective heating surfaces must not exceed the softening temperature of ash, t_i (see Sec. 2.4.). With curtain walls at the furnace outlet, the temperature $\mathbf{0}_{i}^{\sigma}$ is usually taken to be in the range of 1 100-1 200°C and for some clinkering fuels, even in the lower range of 1 000-1 050°C.

7.2. Burners and Their Arrangement

The desired intensity and completeness of pulverized fuel combustion in the furnace space can be achieved through the proper supply and intermixing of pulverized fuel (air-borne dust) with secondary air in a burner assembly (called simply 'burner' in further discussion). Burners do not ignite the fuel. Their function is to prepare two individual flows, a dustair mixture and secondary air, for ignition and active burning in the furnace space. To achieve this, part

of the hot furnace gases should be sucked into the fresh dust-air jet to preheat it and the ignited fuel should be intermixed in due time with the secondary air. For this purpose, hot-air and dust-air flows are introduced into the furnace space at different speeds and with different degrees of turbulization.

There are two main types of burner: the straight-flow burner and the turbulent, or vortex, burner. In a vortex burner, dust-air mixture and secondary air are fed as whirled (turbulized) jets which form a conc-shaped expanding flame in the furnace space (see Fig. 4.10). Vortex burners have a circular cross section. In straight-flow burners, the air-dust mixture and secondary air are blown in as parallel jets. Their intermixing in the furnace space is ensured mainly by an appropriate arrangement of burners on the furnace walls and by providing a particular aerodynamic pattern of jets in the furnace space. Burners of this type may be either circular or rectangular in cross section.

Turbulent burners. The main types of turbulent burners are as follows: (a) two-scroll burners (Fig. 7.4a) in which two scrolls are provided for whirling the dust-air mixture and secondary air; (b) straight-scroll (single-scroll) burners (Fig. 7.4b) in which the dust-air mixture is supplied in a straight flow and spread by a dissector and the secondary air is whirled in a scroll; (c) scroll-vane burners (Fig. 7.4c) with a scroll for the dust-air mixture and an axial vanc-type whirler for the secondary air; and (d) vanetype burners in which the whirling of the dust-air mixture and the secondary air is effected by axial and tangential vanes.

Turbulent burners have a throughput capacity ranging from 1 kg to 3.8 kg reference fuel per second, with their heat power ranging respectively from 25 MW to 100 MW. Two-scroll and scroll-vane burners are more popular, the latter being made for a high heat power (75-100 MW). The

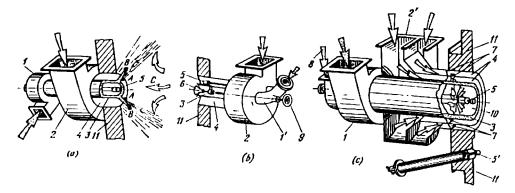


Fig. 7.4. Types of turbulent pulverized-coal burners

(a) two-scroll burner; (b) straight-scroll burner; (c) scroll-vane burner; 1—scroll for fuel-air mixture; 1'—fuel-air mixture inlet pipe; 2—secondary air scroll; 2'—secondary air inlet duct; 3—annular channel for discharging the fuel-air mixture into the furnace; 4—ditto, for secondary air; 5—main fuel oil burner; 5—dissector at fuel-air mixture outlet; 7—swirl vanes for secondary air; 5—admission of ternary air through axial channel; 9—dissector adjustment control; 10—swirer of axial air flow; 11—furnace lining; 4B—boundary of fuel-air mixture ignition; C—suction of furnace gases to flame root

principal aerodynamic characteristic of a burner is the vorticity parameter n (see Sec. 4.4). It ranges from 1.5 to 5, the greater values (n=3.5-5) relating to the whirling of the secondary air flow.

Burners with a higher vorticity parameter give a wider jet with a larger angle of expansion and with a larger zone for recirculation of the hot gases to the flame root, which ensures quicker fuel preheating and ignition. Burners with an elevated value of n are employed for the combustion of low-reactive poorly ignitable fuels (with a relatively low yield of volatiles). In vane-type burners, tilting vanes are sometimes employed, making it possible to control burner aerodynamics.

Among various types of whirlers, profiled vanes have the least resistance for the same vorticity parameter and for that reason are employed widely in powerful burners for whirling the dustair mixture and the secondary air. Burners with dissectors (of the type of Fig. 7.4b) have a lower vorticity parameter n and produce a less expanding jet; they are used in some cases with fuels with a high yield of volatiles, but the dissector may not function reliably under the action of an intensive radiant heat flow.

The completeness of fuel burning heavily depends on the ratio of the axial velocities of the primary and secondary air flows in a burner. The velocity of the primary flow (dust-air mixture) is usually $w_1 = 16-25$ m/s, higher values being typical of powerful burners. The optimal velocity of secondary air is $w_2 = (1.3-1.4) w_1$.

Turbulent burners can be used with any kind of solid fuel, but are used more widely for low-volatile grades. Burners of elevated heat power are provided with two controlled coaxial channels for secondary air (see Fig. 7.4c), which make it possible to maintain the required air velocities at reduced loads: at loads less than 70% of the rated value, the peripheral channel is closed.

Straight-flow burners. Burners of this type turbulize the air flows less substantially than do turbulent burners and produce a long-ranging jet with a low expansion angle and weak intermixing of the primary and secondary flow. Efficient combustion is achieved by making the jets from various burners interact with one another in the furnace space. Straight-flow burners may be either fixed or tiltable which facilitates combustion control (Fig. 7.5a). Burners of rectangu-

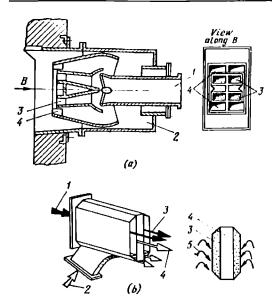


Fig. 7.5. Straight-flow burners for pulverized coal

(a) with tilting nozzle at the exit of fuel-air mixture; (b) with central channel for hot air; 1-supply of fuel-air mixture; 2supply of hot air; 3-fuel-air mixture outlet; 4-hot mir outlet; 5-suction of furmace gases

lar shape, especially those extended along the height, are characterized by a high injection of the surrounding gaseous medium into the jet sides. For that reason, such burners with the external mixing of dust-air flow (Fig. 7.5b) possess certain advantages over those with internal mixing, particularly in that they ensure easier ignition. Straight-flow burners are, as a rule, of a relatively low throughput capacity; in high-capacity boilers, a number of such burners are combined

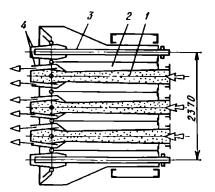


Fig. 7.6. An assembly of three straight-flow pulverized-coal burners

1—supply of fuel-air mixture to burner; 2—secondary air supply to burner; 3—pipe for mounting the firing fuel-oil burner with electric gas igniter; 4—tilting air pipe

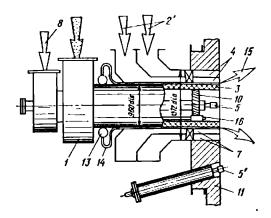
in a burner assembly (Fig. 7.6). Straight-flow burners are mainly employed with high-reactive fuels: brown coals, peat, oil shales and high-volatile coals. The velocity of the dust-air mixture at the burner outlet is taken as $w_1 = 20-28$ m/s and the optimal velocity of secondary air, $w_2 = 1.5-1.7$ of w_1 .

Combined burners. In some cases. power stations have to operate, simultaneously or alternately, on a variety of fuels, and must employ combined burners which can efficiently burn each kind of fuel. As an example, Fig. 7.7 shows a combined burner in a high-capacity steam boiler which can operate on three kinds of fuel: solid (basic), fuel oil and natural gas. The burner has an enlarged diameter in the central channel in which the main fuel oil burner and a whirling device for turbulizing the axial air flow are arranged. Natural gas is fed through feeder tubes in thin jets into the space between the turbulized axial and secondary air flows ensuring thorough intermixing and stable burning of fuel.

Burner arrangement. Burners are arranged on the walls of a furnace so as to ensure as complete fuel combus-

Fig. 7.7. Schematic of a burner for three kinds of fuel

Numbering as in Fig. 7.4; additionally: I3—annular duct for natural gas; I4—pipes for admitting natural gas to burner, arranged around the primary air channel 3; I3—natural gas to furnace; I6—electric gas lenter



tion in the flame core as possible, provide favourable conditions for the removal of slag from the furnace (in either dry or molten state), and avoid clinkering of the furnace walls. The optimal arrangement of burners is found by properly considering their type and characteristics. For instance, turbulent burners give a shorter and wider flame than do the straight-flow type. The primary and secondary air flows are intensively intermixed due to the energy of turbulent motion, which ensures a thorough burning-off of fuel in the flame core (up to 90-95%).

The main parameter of turbulent burners is the diameter of their port, or embrasure, D_p. Burners are arranged

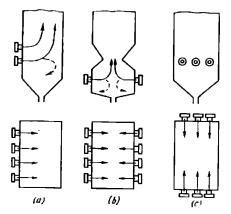


Fig. 7.8. Arrangement of pulverized-coal turbulent burners on furnace walls

(a) front arrangement; (b) double-front (opposite) arrangement; (c) opposite arrangement on side walls

at a distance of $(2.2-3) D_p$ from one another and at a distance of $(1.6-2) D_p$ from the side walls of the furnace, which helps to prevent the flames from interacting prematurely and touching the walls [57].

Some typical arrangements of turbulent pulverized-coal burners in the furnace are illustrated in Fig. 7.8. The front and double-front arrangements (Fig. 7.8a and b) may have one or two rows (tiers) of burners along the height. With a single-front arrangement, the rear water wall of the furnace has an elevated heat absorption (10-20% above the average value), so that the furnace depth (in a dry-bottom furnace) must be not less than $b = (6-7) D_p$ in order to keep the rear wall from clinkering. The opposite double-front arrangement of burners is typical of high-capacity steam boilers where the required number of burners cannot be arranged on a single front wall even in two tiers.

With the opposite arrangement of burners, heat absorption by water walls in a furnace is much more uniform. The opposite burner arrangement is employed primarily in slagging-bottom furnaces, since colliding flames are deflected both upwards and downwards and thus can maintain a high temperature level at the furnace bottom. Efficient interaction of opposite flames is achieved if the furnace width is $b = (5-6)D_p$. In low-capacity boilers, burners are usually arranged

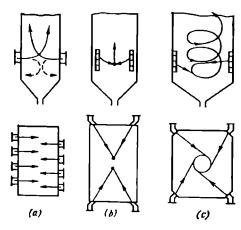


Fig. 7.9. Arrangement of straight-flow pulverized-coal burners on furnace walls
(a) opposite displaced arrangement; (b) corner arrangement with encountering jets (block arrangement); (c) corner arrangement with tangential jets (tangential arrangement)

opposite each other in a single tier on the side walls of the furnace (Fig. 7.8c); the depth of the furnace is then determined only by their arrangement. In this scheme, an elevated temperature of gases is obtained in the middle portion of the furnace across its width. Typical arrangements of straightflow burners are illustrated in Fig. 7.9. Burners of this type can ensure complete combustion of fuel only by the turbulization of the flames from individual burners upon their collision in the furnace space. All the arrangements shown in the figure are widely employed for the combustion of peat, brown coals and young coals.

In the burner arrangement with opposite displaced jets, developed at the Moscow power engineering institute, peat and brown coal can be effectively burned as a result of the intensive turbulization of the flame in the main combustion zone. This is achieved by forming a large velocity gradient between adjacent jets moving in opposite directions.

A burner arrangement with corner burners and flame jets directed tangentially to an imaginary circle 1-2.5 m in diameter in the furnace centre (Fig. 7.9c) has found wide application in many types of steam boilers, including high-capacity units (Fig. 7.10). This arrangement is advantageously characterized by the uniform distribu-

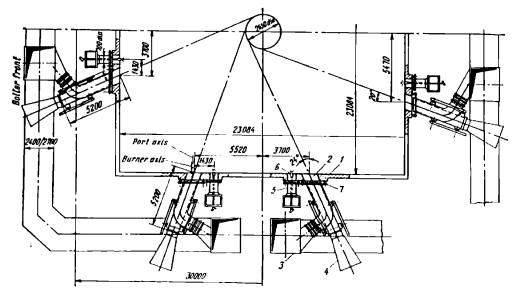


Fig. 7.10. Arrangement of straight-flow !burners in the furnace of P-67 boiler of 800-MW monobloc unit fired with Berezovo brown coal '

4-front wall of furnace; 2--straight-flow hurner port; 3-hot (secondary) air duct; 4-fuel-air mixture channel; 5-firing fuel-oil burner; 6-head of fuel-oil burner; 7-temperature compensator

tion of heat flows between the furnace walls and by the low probability of clinkering of the walls since they are in contact with furnace gases which have already been cooled appreciably. When applied in slagging-bottom furnaces, this arrangement causes molten slag droplets to settle on the walls of the primary furnace and thus increases the fraction of ash transferred to slag.

An arrangement with corner burners and encountering jets (block arrangement, Fig. 7.9b) is employed for coal combustion. In this arrangement, the flame core is highly turbulized. Its drawback is the likelihood of clinkering of the front and rear furnace walls if the flame moves from the furnace centre (the zone of relatively high pressure) to both sido walls.

A tangential burner arrangement can be employed in furnaces with a nearly square cross section, i.e. with the a/b ratio equal to 1-1.2. This ensures good aerodynamics in the furnace space.

7.3. Dry-bottom Furnaces

Dry-bottom furnaces, i.e. those in which slag is removed in the solid state, are mostly of the open type—with a constant cross section along their height. Their design is determined by the flame's pattern of motion which can be straight, vertically turbulized or horizontally turbulized (Fig. 7.11). Dry-bottom furnaces are characterized by the provision of a dry-bottom hopper which is formed in the lower portion of the furnace by inclining the front and rear water walls at an angle of 50-60° so that the distance between them decreases at the

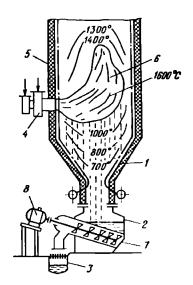
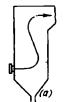


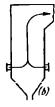
Fig. 7.12. Dry-bottom furnace 1—dry hottom hopper: 2—slag bath with water; 3—hydraulic ash disposal channel; 4—hurner; 5—water walls; 6—flame core; 7—slag-removing screw mehanism; 8—electric motor

bottom to b' = 1-1.2 m. The drybottom hopper intensively cools the furnace gases in the furnace bottom, so that molten slag particles which enter this zone are cooled quickly, solidify, and fall along the hopper sides into a slag pit (Fig. 7.12). The quantity of ash collected in this way in the dry-bottom hopper is not high, usually 5-10% of the total ash content of the fuel. Granulated slag particles are continuously removed from the bath by a screw, paddle or rotor device. The water bath also serves as the hydraulic seal preventing the suction of cold air from boneath into the furnace. The aerodynamics of the furnace space should be such that the gas temperature near the water walls be

Fig. 7.11. Patterns of flame motion in dry-bottom furnaces

(a) straight-flow S-shaped flame; (b) opposite straight-flow flame; (c) vertically turbulized flame; (d) combination of straight-flow and horizontally turbulized flame









below the temperature t_1 (see Sec. 2.3) at which ash particles become sticky and can cause clinkering of the water walls. For this reason, the average heat release rates (per m² of furnace cross section and per m³ of furnace volume) of dry-bottom furnaces are usually not high $(q_f = 3-4 \text{ MW/m²}, q_V = 100-140 \text{ kW/m³})$. This inevitably results in large dimensions of dry-bottom furnaces, and therefore, a high consumption of metal.

For instance, the furnace of a once-through boiler type P-59 of a 300-MW monobloc unit, when designed for the combustion of Moscow district brown coal with the straight-flow flame as shown in Fig. 7.11b, has the dimensions $a \times b \times h_f = 21.8$ m \times 9.56 m \times 48 m.

Dry-bottom furnaces are usually employed for the combustion of pulverized solid fuels with a high or moderate yield of volatiles ($V^c > 25\%$).

The most popular combustion scheme in dry-bottom furnaces is that with a rising straight-flow flame (Fig. 7.11a and b), which is formed by turbulent burners arranged on the front wall or by straight-flow burners arranged on two walls (opposite arrangement). To fire high-capacity boilers on Siberian brown coals, another combustion schome is proferred (see Fig. 7.10) in which a vertical turbulized flame is formed by straight-flow burners arranged in several tiers along the furnace height (see Fig. 7.9c). This scheme decreases the probability of the flame touching the furnace walls and the associated risk of clinkering, while the distribution of the burners along the furnace height (on a length of up to 12 m) results in less heat released at each tier, thus lowering the temperature level in the extended flame core and noticeably diminishing the formation of harmful nitrogen oxides NO_r. Furnaces with a horizontally turbulized flame, designed by Prof. V. V. Pomerantsev, successfully operate on milled peat and brown coals (Fig. 7.11d). Fine fractions of fuel burn up in the straight portion of the flame, while the coarser particles are thrown to the bottom, entrained by the secondary air jet, and involved into vortex motion until completely burnt off.

In dry-bottom furnaces, virtually complete combustion is achieved at an excess air ratio at the furnace outlet $\alpha_f = 1.15 - 1.20$. Because of the inevitable suction of cold air from the outside ($\Delta \alpha_f = 0.05 - 0.1$), the excess air ratio in burners is somewhat lower: $\alpha_b = \alpha_f - \Delta \alpha_f = 1.05 - 1.1$.

7.4. Slagging-bottom Furnaces

To remove slag in the molten state, the temperature of gases at the walls in the lower portion of a furnace should be higher than the temperature of slag fluidity, i.e. $\vartheta_g > t_{fl}$, where $t_{fl} = t_3 + (50-100)^{\circ}$ C is the temperature of the normal fluid state of slag. Such conditions at the furnace bottom can be provided by moving the flame core closer to the furnace bottom and by protecting the water walls in that zone with carborundum refractories (facing of water wall tubes). For better attachment of the facing, pins 10-12 mm in diameter and 12-15 mm long are first welded onto the tube surface on the fire side, then the refractory facing is applied (Fig. 7.13). In the design of heat-insulated water

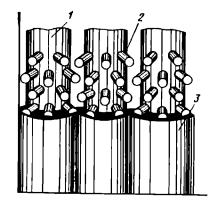


Fig. 7.13. Portion of refractory-faced water wall

1—water wall tube; 2—pins before facing; 3—
refractory facing

walls proposed at the Ordzhonikidze machine-building works in Podolsk, helically finned tubes are used instead of pinned tubes.

The bottom of a slagging-bottom furnace is either horizontal or slightly inclined towards the furnace centre. The tubes of the bottom are covered by two or three courses of refractory bricks laid on refractory mortar. One or two refractory-lined holes 500 mm × 800 mm (slagholes) are left in the bottom centre to pour off molten slag. Thin jets of molten slag flow over the edges of the slagholes into the slag bath where they solidify on contact with water.

The fraction of the total ash removed to slag in slagging-bottom furnaces is substantially higher than that in drybottom furnaces: $a_{sl} = 0.2\text{-}0.4$. Solidified slag is removed continuously from the slag bath by paddle, screw or rotor conveyers.

Slagging-bottom furnaces may have a single shaft (open and semi-open furnace) or two or three shafts. Combustion can be organized in a straightflow flame, intersecting flame jets or by the cyclone principle.

The simplest type of slagging-bottom furnace is a furnace with a single

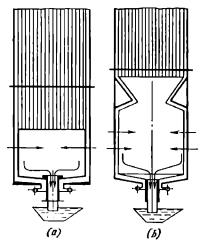


Fig. 7.14. Slagging-bottom furnaces with straight-flow flame
(a) straight-wall (open-shaft) furnace; (b) constric-

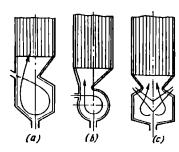


Fig. 7.15. Turbulent furnaces with intersecting jets

(a) designed at the Moscow power engineering institute; (b) designed at the Central boiler and turbine institute; (c) gamma-furnace, designed at the All-Union heat engineering institute

shaft and straight-flow flame (Fig. 7.14a). The lower portion of the water walls and the furnace bottom are refractory-faced to form a zone of elevated gas temperature (slag-melting zone). In that case, turbulent burners are employed and arranged at a lower height above the bottom. Since, however, much heat is given up to the upper zone, the furnace has limited possibilities for combustion control: at loads below 0.7-0.8 of the rated value, slag may start solidifying, first on the walls and then on the bottom. Also, the fraction of total ash removed to slag in an open-type furnace is not high: $a_{nl} = 0.1-0.15$.

Constructing two opposite walls in a furnace ensures the isolation of a combustion chamber, or primary furnaco within that furnace (Fig. 7.14b). This substantially reduces the heat transfer into the upper zone, making it possible to appreciably raise the temperature of gases in the primary furnace (up to 1600-1800°C). The heat release rate in the primary furnace may be as high as $q_V^{pr} = 500$ -800 kW/m³, and the fraction of total ash removed to slag: $a_{sl} = 0.2-0.4$. The boiler can operate in a wider range of loads with slags being removed in the molten state.

In a furnace with intersecting flame jets (Fig. 7.15), the primary furnace is isolated by constructing one or two walls. Straight-flow burners are arran-

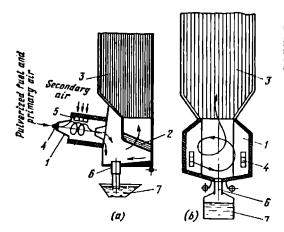


Fig. 7.16. Cyclone furnaces

(a) furnace with horizontal cyclone-type primary furnace; (b) bottom-type primary furnace open at the top; 1-primary furnace (cyclone); 2- slag-catching tube hundle; 3-cooling chamber; 4-burner; 5-secondary air nozzics; 6-slag-hole; 7-slag-bath

ged so as to turbulize the flame around a horizontal axis. The flame makes a full turn in the zone of refractory-faced water walls, after which hot gases pass in interspaces between the burners, intersect the jets of fresh dust-air mixture and thus ensure their quick heating and stable burning. The organized motion of gases along the walls and bottom of the furnace provides favourable conditions for the stable removal of molten slag even at boiler loads 40-50% of the rated value. The heat release rate of the primary furnace is 500-600 kW/m³.

A more efficient separation between the zones of combustion and gas cooling is attained in furnaces provided with cyclone-type primary furnaces (Fig. 7.16). These furnaces may be classed with two-shaft furnaces. The cyclone method of combustion consists essentially in the flame in the primary furnace being whirled by either tangential high-speed jets of secondary air (80-120 m/s) or tangential dust-air jets from burners (see Fig. 3.1). The entire surface of the primary furnace is covered by water walls made from refractory-faced pinned tubes. Fuel particles in the primary furnace are acted upon by two forces: the centrifugal force which throws them onto the walls of the primary furnace and the aerodynamic force which tends to carry off fuel particles and gases from the primary

furnace. The ratio of these forces depends on particle size, because of which fuel particles are unevenly distributed in the primary furnace: coarser particles are thrown against the walls and become involved in the vortex motion until they burn off completely, while finer fractions burn in the centre. A cyclone-type primary furnace can operate on coarse pulverized fuel and in some cases (horizontal cyclones), on crushed fuel, which can thus reduce the cost of fuel pulverization. Intensive vortex motion also ensures that a large fraction of ash be removed to slag (a_{sl} up to 0.6-0.85), the greater value relating to horizontal cyclone-type primary furnaces.

Horizontal cyclone-type primary furnaces (Fig. 7.16a) are made to a diameter of 1.8-4 m. The length of the cyclone is 1.2-1.3 times its diameter. The heat power of a cyclone may be as high as 150-400 MW. The heat release rate in cyclones is rather high $(a_v = 2-6 \text{ MW/m}^3)$ at the gas temperature 1 800-1 900°C and excess air ratio $\alpha_r = 1.05-1.1$. Since, however, an extended shaft is needed to cool the gases, the total heat release rate of furnaces with horizontal cyclones does not exceed 200-300 kW/m³, i.e. it is only slightly higher than that of the conventional single-shaft slagging-bottom furnaces.

High speeds of the secondary air are attained by using special-type blo-

wers which can develop a head of 10-20 kPa (1 000-2 000 mm H₂O), i.e. 2-3 times the common air head. The design of furnaces with cyclone-type primary furnaces is more intricate than that of conventional single-shaft furnaces and their cost is correspondingly

higher.

In the slagging-bottom furnace design developed at the Barnaul boiler works, a vertical primary furnace in which gases exit from the top is arranged below the cooling chamber (Fig. 7.16b). The primary furnace is octahedral in shape and is assembled from plane water walls connected to the circulation circuit of the water walls of the main furnace. This furnace design is much less expensive to implement than that with a horizontal cyclone. Usually two primary furnaces are provided for a single cooling chamber. Straight-flow slit-type burners are arranged on the four walls of the primary furnace; the jets are directed tangentially, with the velocities of primary and secondary air being in the common ranges $(w_1 =$ = 25-35 m/s, $w_2 = 40-50$ m/s). All water walls of the primary furnace are refractory-faced.

The advantages of slagging-bottom furnaces over the dry-bottom type are essentially as follows. When burning the same kind of fuel, the heat loss with unburned carbon, q_4 , in a slagging-bottom furnace can be reduced roughly by 30%. The total heat release rate per unit furnace volume turns out to be 20% higher on the average, which means that the slagging-bottom furnace can be made smaller. Due to better tightening of the bottom portion, air suction into the furnace decreases, with the result that heat loss with waste gases is somewhat lower. Furthermore, slagging-bottom furnaces are characterized by lower expenditures on ash disposal.

On the other hand, slagging-bottom furnaces possess certain drawbacks. For instance, since more ash is removed as high-temperature slag, the heat loss with slags, q_s , is higher and in many cases can overbalance the reduction in q₄. In single-shaft slaggingbottom furnaces, the possibilities of boiler load control are restricted by the conditions of slag removal (which must remain liquid). A higher temporature level in the flame core results in a higher yield of harmful nitrogen oxides. Therefore, the choice between dry-bottom and slagging-bottom furnaces for a particular kind of fuel should be made by properly considering all their advantages and drawbacks. One should also take into account that not all kinds of fuel can be burned in slagging-bottom furnaces. Those which form low-fusible ash $(t_{s} = 1 150-1 300^{\circ}C)$ cause no difficulties, while for fuels with $t_3 >$ > 1 350°C the conditions for the formation of molten slag should be properly calculated.

Slagging-bottom furnaces are advantageous for the combustion of lowreactive fuels (anthracite, semianthracite, lean coals), when they ensure a noticeable reduction of the heat loss with unburned carbon, and of fuels with a low temperature of ash fusion which might cause strong clinkering of the water walls in dry-bottom fur-

naces.



GAS AND FUEL OIL-FIRED FURNACES

8.1. Furnace Design

The conditions of combustion of natural gas and fuel oil have much in common, and therefore, both fuels can be burned in furnaces of the same design. In most cases, such furnaces are designed primarily for fuel oil with natural gas as the auxiliary fuel. The combustion characteristics of fuel oil and natural gas are similar in the following respects:

- 1. Both fuels contain practically no adventitious moisture and form roughly the same volumes of combustion products; therefore, the blowers of a steam boiler can efficiently operate irrespective of whether fuel oil or natural gas is being burned in the boiler furnace.
- 2. Burning of fuel oil and natural gas occurs in the vaporized state (homogeneous medium) following the laws of branched chain reactions (see Sec. 4.2). The intensity of burning in both cases is determined by the conditions of intermixing, and the highest allowable heat release rates of the furnace volume are rather close to each other

(300 kW/m³ for fuel oil and 350 kW/m³ for natural gas). Thus, for the same steam output of a boilor, the furnace dimensions for these two kinds of fuel can be taken to be practically the same.

3. Both fuels form almost no ash on combustion (the ash content of fuel oil A^d is less than 0.3%), which avoids clinkering of the water walls in the furnace and makes slag-handling facilities unnecessary. In view of this, furnaces for both fuels are made with a horizontal or slightly inclined bottom, with man-holes provided for repairs (Fig. 8.1).

4. Since the fuel is in the gaseous (or vaporized) state, it can be more easily intermixed with air, which ensures virtually complete combustion at a high heat release rate and low excess air ratio $\alpha_c = 1.02-1.05$. For both fuels, air can be preheated to the same temperature $(t_{ha} = 250-300^{\circ}\text{C})$, which makes it possible to employ combined gas-fuel oil burners with close values of the volume flow rate of air and almost the same resistance.

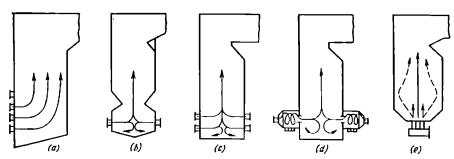


Fig. 8.1. Types of fuel oil-fired furnaces

(a) open-type furnace with single-front multi-tier arrangement of burners; (b) furnace with constriction and opposite (double-front) burner arrangement; (c) open-type furnace with opposite double-tier burner arrangement; (d) furnace with opposite cyclone primary furnaces; (e) furnace with straight-flow or turbulent bottom burners (dotted lines)

With intensive burning, both fuels form a relatively short flame core zone near the burners. In fuel oil combustion, this zone is characterized by a rather high temperature level and an extremely intense heat flux radiated onto the water walls. This may load to the overheating of the tube metal and the appearance of high-temperature corrosion, resulting in a high concentration of nitrogon oxides in the flame core.

In the vertical section, gas and fuel oil furnaces may be open, with restriction or with cyclone primary furnaces (Fig. 8.1). Most industrially made gas and fuel oil-fired boilers are provided with conventional furnaces of prismatic shape with a single-front or double-front (opposite) arrangement of burners. In the single-front arrangement, burners are mounted in several (three or four) tiers. This arrangement is less expensive and more convenient in operation, but cannot ensure uniform filling of the furnace space by the flame and is inapplicable in furnaces of a short depth (less than 6 m), since the temperature and heat absorption of the rear water wall would then increase intolerably.

In an opposite arrangement of burners, the water walls of the furnace operate under more favourable conditions. The flame is concentrated in the central high-temperature zone of the furnace space. The opposite motion of the flames creates unfavourable conditions for turbulization and better fuel burn-up in the tail portions of the flames and, under identical conditions, results in an increase in the heat release rate in the flame core zone by 20-30%. A constriction in the furnace can increase flow turbulization in the flame core zone and in the zone where fuel afterburning occurs at the outlet from the combustion chamber.

In an experimental series of steam boilers for 300-MW monobloc units, a new proposal is to organize combustion in opposite cyclone-type primary furnaces in order to decrease the intensity of heat flows onto the furnace water walls (Fig. 8.1d). The high turbulization of flow in the cyclone primary furnace enables 85-90% of the fuel to be burned. Cyclone primary furnaces are covered with pinfinned tubes with carborundum refractory facing. This design, howover, leads to a higher temperature of the flame and higher heat flow onto the water walls, making it not the best solution for some kinds of fuel.

Gaseous fuels are known to produce flames of a lower emissivity than fuel oil, so that when a boiler is changed from fuel oil to natural gas firing, heat absorption by the furnace space decreases, while the temperature of combustion products at the outlet from the furnace becomes higher. In open furnaces at the rated load, this temperature difference may be as high as 100 deg C, which inevitably changes the temperature conditions of the subsequent heating surfaces, primarily that of the superheater. In open furnaces with multi-tier single-front burner arrangement, the temperature of gases at the furnace outlet is equalized in such cases by changing the pattern of the flame core: when burning natural gas, only two or three lowermost burner tiers are in operation; for a change from gas to fuel oil, burners of the upper tiers are fired; in later designs, the gases are recirculated into the furnace to accomplish the same purpose.

Recently, it has been proposed to diminish local heat flows on furnace water walls by arranging the burners in the bottom of an open furnace and by controlling the whirling of the secondary air flow (Fig. 8.1e). In fuel oil combustion, the degree of whirling is decreased, so that the flame extends to a greater height in the furnace and thus noticeably decreases local heat flows onto the water walls, while the temperature of gases at the furnace outlet rises substantially. When burning natural gas, the degree of whirling is increased to make the flame wider and shorter.

8.2. Fuel Oil Burners

Gas and fuel oil-fired boilers are provided with combined burners since the volume flow rates of air for the combustion of natural gas and fuel oil are practically the same. The type of burner is selected depending on the heat power of the boiler and on the burner arrangement on the furnace walls, so as to ensure the best filling of the furnace space by the flame. For more efficient combustion, fuel oil is atomized in nozzles (see Sec. 4.2). A fuel oil burner comprises a nozzle, a channel to supply hot air, and a device for air whirling (air register). Depending on the method of atomization, burners are classified as mechanical, steam-mechanical or steam atomizer burners [13].

Mechanical burners are the most popular type of fuel oil burner. Atomization is effected by supplying fuel oil under an excessive pressure (2.5-4.5 MPa) into the whirling chamber and ejecting it through a narrow hole, or nozzle (Fig. 8.2a). Fuel oil enters the whirling chamber through a number of tangential channels and is turbulized into an intensively rotating whirl with the potential velocity distribution:

$$w_t r = w_0 R \tag{8.1}$$

where w_i is the tangential velocity at a distance r from the axis of the outlet section of the nozzle, w_0 is the velocity at the inlet to the tangential chan-

nel, and R is the distance between the axis of the burner and that of the tangential channel.

The whirled flow has a reduced pressure in the core which is filled with a gas whirl, so that the fuel flows from the nozzle through an annular section of a thickness $r_0 - r_{ini}$, where r_0 is the nozzle radius and r_{ini} is the radius of the internal gas whirl. Depending on the ratio of the tangential and axial velocity components, the fuel film at the burner outlet expands at an angle φ and is disintegrated by the oncoming air flow into numerous droplets which move along a parabolic path.

The theory of centrifugal burners has been developed by G. N. Abramovich. The principal characteristics of a burner are the dimensionless geometric parameter:

$$A = Rr_0/r_{inl}^2 \tag{8.2}$$

which is determined by the design characteristics of a burner (see Fig. 8.2); the coefficient of the free crosssectional area

$$\xi = 1 - (r_{inl}/r_0)^2 \quad (8.3)$$

the flow rate coefficient μ which correlates the total pressure of the fuel at the inlet to the centrifugal chamber with the energy of the outcoming flow, and the jet expansion angle ϕ (see Fig. 8.2b).

The throughput capacity of a mechanical centrifugal burner, B_m , kg/s, depends in the first place on the cross-sectional area of the ejecting nozzle,

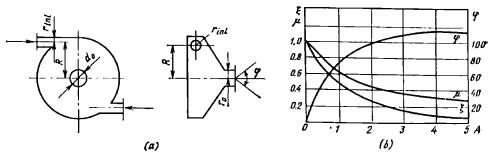


Fig. 8.2. Principal diagram and main characteristics of a centrifugal mechanical burner (a) principal diagram; (b) effect of the geometric characteristic of a burner (A) on the free area coefficient (ξ), flow rate coefficient (μ), and jet expansion angle (ψ)

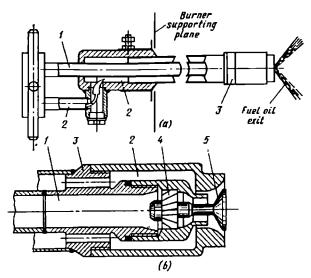


Fig. 8.3. Steam atomizer fuel-oil burner type TKZ-4

(a) schematic; (b) longitudinal section through burner head; 1-fuel oil inlet; 2-steam inlet;

5 -head housing; 4- whirler; 5-dissector

 f_0 , m², and the fuel pressure p_m , MPa:

$$B_m = b\mu f_0 \sqrt{p_m \rho_m} \qquad (8.4)$$

where b is in addition a numerical factor and ρ_m is the fuel oil density, kg/m^3 .

The throughput capacity of powerful centrifugal burners with the nozzle diameter $d_0 = 4.10$ mm may range from 0.83 kg/s to 4.5 kg/s (3.16 t/h). The burner capacity can be controlled within 60-100% of the rated value by lowering the fuel oil pressure.

Steam-mechanical burners have a wider range of capacity control since at lower loads the fuel oil is atomized by the energy of steam. A burner of this type is essentially a common mechanical oil burner provided with an additional annular channel to supply steam at the pressure $p_{st} =$ = 0.2-0.4 MPa (Fig. 8.3). In the burner design shown in the figure, the centrifugal whirling chamber is replaced by an axial whirling device with a conical dissector. The steam flow, whose speed is near the critical value, penetrates the fuel oil iet and disintegrates it into fine droplets. The flow rate of steam for atomization is not more than 10% of that of fuel oil. The burner capacity can be controlled within 20-100% of the rated value.

Multi-nozzle steam-mechanical oil burners, which form a flat flame, are also in use (Fig. 8.4).

In solid-fuel fired boilers, steam atomizer oil burners are employed only at boiler start-up. They are inefficient in long-term operation be-

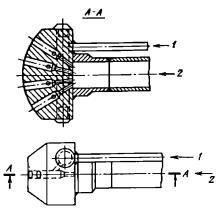


Fig. 8.4. Head of a flat-flame burner I—fuel oil inlet; 2 - steam inlet

cause of the high consumption of steam for atomization (30-40% of fuel oil consumption). Steam is ejected at a pressure of 0.4-0.6 MPa and disintegrates the fuel oil jet into fine droplets. The fuel oil can be supplied at a low pressure. Burners of this type are simple in design and ensure fine atomization of fuel oil even at a rather low preheating temperature (up to 80°C). Starting-up burners can usually cover up to 30% of the rated boiler capacity.

Due to centrifugal force, fuel oil is ejected from the nozzle of a mechanical burner as a thin hollow cone. The thickness of the initial oil film is $\delta_f = 0.56 \mu r_0$, which is 0.5-2 mm. The film then expands and breaks into fine droplets, the largest of which are roughly equal to the film thickness in diameter.

Oil droplets move at a speed of 60-80 m/s and are further atomized by the dynamic pressure of the oncoming gaseous medium (air or steam). This process is called secondary atomization.

The average diameter of oil droplets in mechanical burners is roughly $\delta_{a\,\nu}=300~\mu m$; it depends proportionally on $d_m^{0.5}$ and in inverse proportion on $p_m^{0.25}$. As may be seen, oil pressure has only a slight effect on the size of the droplets. The burner nozzle diameter d_0 has a somewhat greater influence on the fineness of atomization.

In steam-mechanical burners, $\delta_{av} = 50{\text -}100 \,\mu\text{m}$ and depends on the energy of supplied steam and the efficiency of its utilization for [uel oil atomization.

In powerful mechanical burners, the largest oil droplets may be up to 1.5-2 mm in size. The time for their evaporation and burning is roughly 2 s. Hence, the heat release rate per unit volume of the furnace must not be more than 200-250 kW/m³ so as to ensure complete fuel combustion (see Fig. 7.2). It should also be taken into consideration that larger droplets move in an ascending

flow more slowly than the surrounding gases, and therefore, the timeof their presence in the furnace is somewhat longer. With the help of steam-mechanical burners, the heat release rate of the furnace can be increased.

Controlling the burner throughput capacity. The flow rate of fuel oil supplied for combustion can be controlled by two methods: by varying the fuel oil pressure in all operating burners (qualitative method) or by switching some of the burners on and off (quantitative method).

As follows from formula (8.4), the first method requires substantial reduction of the initial pressure, since B_m is proportional to $p_m^{\bullet,\bullet}$. For instance, to diminish the throughput capacity by 40% (from 100% to 60% of the rated value), the fuel oil pressure must be reduced to 0.37 of the initial value. On the other hand, a sharp pressure reduction in mechanical burners is inadmissible, since a lower intensity of whirling will result in a thicker oil film, lower velocity at the nozzle outlet, and larger droplets. An increase in the initial pressure of fuel oil will require more intricate and expensive equipment for oil transport and flow rate control and higher operating expenditures. For these reasons, the throughput capacity of mechanical burners is controlled by using the combined qualitative-quantitative method.

When the number of oil burners in a furnace is sufficiently high, the load on the boiler can be diminished by switching off some of the burners in a group. In such a case, the pressure of fuel oil supplied to the remaining burners increases, allowing the boiler load to be further controlled by varying the fuel oil pressure.

In high-capacity steam boilers, steam-mechanical burners are more popular, as they ensure proper control in the whole range of operating loads. Atomization of fuel oil can be improved by preheating the fuel to a higher temperature (see Sec. 2.4).

8.3. Combustion of Natural Gas

An essential feature of natural gas combustion is that the mixture is formed from sharply different volumes of gas and air: combustion of 1 m³ of natural gas requires roughly 20 m³ of hot air. For this reason, the gas supply channel in a burner has a rather small cross-sectional area. Proper intermixing of the gas and air can be ensured only by introducing the gas into the air flow in the form of thin jets of a high penetrability (the gas velocity is up to 120 m/s while the velocity of primary air is only 25-40 m/s). Gas burners belong to the class of burners with partial internal mixing, since complete intermixing of gas and air is not effected in the burner proper, but is attained in the furnace space. As a result, part of the gas in the high-temperature zones, where oxygen may be deficient, is subjected to thermal dissociation (pyrolysis) with the formation of soot particles. This is why a gas burner, like an oil burner, forms a sufficiently bright flame, with the highest combustion temperature appearing at a certain distance from the burner port, which diminishes the risk of burner burn-up.

In most cases, gas is introduced at right angles to the air flow or, in some cases, at a certain angle α to the air flow lines (Fig. 8.5). For a

uniform distribution of gas in the volume of air, the depth of penetration of the individual gas jets must be different. The penetration depth h is the decisive characteristic in calculations; it is determined by the ratio between the momenta of a gas jet and air flow and characterizes the distance along the normal from the jet root to the point where its direction becomes coincident with that of the air flow [26].

The depth of jet penetration is determined mainly by the diameter of a jet and by the ratio of gas and air velocities. With gas holes being oriented along the air flow and arranged in two or three rows, a uniform distribution of the gas in air flow can be achieved by decreasing the diameter of holes in the direction of air motion (Fig. 8.5c).

The principal parameters of a gas burner are as follows:

- 1. The relative length of the internal mixing zone $l_m = L_m/D_a$, where D_a is the diameter of the outlet port of the burner (Fig. 8.6); it characterizes the conditions of partial internal mixing of flows.
- 2. The aerodynamic parameter which characterizes the intensity of turbulent mixing:

$$n_a = \frac{\rho_a w_{wh}^3}{\rho_g w_g^2} \tag{8.5}$$

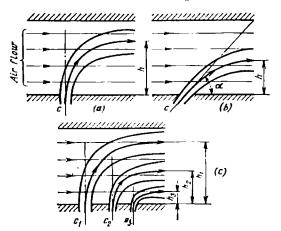


Fig. 8.5. Penetration of gas jet into drifting air flow

(a) single jet entering an air flow at right angles; (b) ditto, entering at an

(a) single jet entering an air flow at right angles; (b) ditto, entering at an angle $\alpha < 90^\circ$; (c) penetration of multiple-row gas jets into an air flow; h—depth of jet penetration

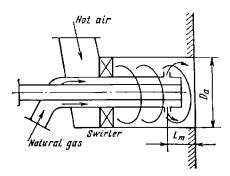


Fig. 8.6. Gas burner with central gas supply

where w_{wh} is the actual velocity of the whirled air flow which is roughly twice the axial velocity (flow rate); $w_{wh} = 2w_a$; the optimal range of n_a is $n_a = 1-1.2$.

3. The dimensions, shape and arrangement of gas ports, which depend on the general design of the burner.

Gas jets can be introduced into the air flow in burners in a peripheral, central or two-sided mode. Figure 8.6 shows the central supply of gas jets. A two-sided gas supply (from the periphery and from a central gas header) is employed in high-capacity gas burners (see Fig. 8.9).

8.4. Combined Gas-fuel Oil Burners

Among the advantages of combined gas-fuel oil burners (gas-oil combustion burners) is that the change from one fuel to the other can be done quite easily. Furthermore, both fuels can be burned under almost optimal conditions.

The air channels in such a burner are common for both fuels, and each type of burner is arranged so as to ensure quick and thorough intermixing of fuel and air. For efficient intermixing with the fuel, the air flow is strongly turbulized in the burner by means of an air-guiding device called an air register.

Air registers may be of the scroll type or with axial or tangential vanes (Fig. 8.7). Scroll-type air registers are too bulky because of the large volumes of air used for combustion, and are employed only in low-capacity burners. Air registers with axial vanes are the simplest in design and have the least hydraulic resistance, but the diameter of the channel must be quite large to accommodate the entire air flow. Tangential vane registers have a somewhat higher resistance than the axial-vane ones, but their passage section can be easily varied at load variations by moving a regulating disc along the burner axis (Fig. 8.8).

For the most part, high-capacity boilers employ gas-oil combination burners of the three types shown in Figs 8.8-8.10. These burners differ from one another in the method of gas introduction into the air flow and in the method of gas flow rate control at varying loads.

A coaxial gas-oil combination burner with a central gas supply is shown in Fig. 8.8. Natural gas flows from the central annular header through two rows of holes of different diameter. Air is supplied through a tangential vane register. The air flow rate is controlled by a moving disc valve. Thus, at a reduced boiler load, the air flow rate decreases maintaining the original whirling intensity and the necessary conditions for fuel-air intermixing. Fuel oil is atomized in a mechanical burner mounted in the

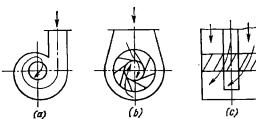
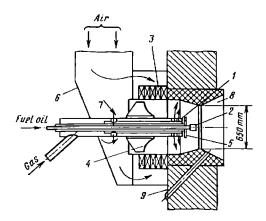


Fig. 8.7. Air registers
(a) scroll type; (b) tangential vane-type;
(c) axial vane-type

Fig. 8.8. Coaxial gas-fuel oil burner with central gas supply

1-annular gas channel; 2-fuel oil burner; 3-tangential vanes; 4-air control gate valve; 5-flame-protecting disc of gas head; 6-air box; 7-air supply to cool the head and disc; 8-conical port; 9-igniter channel



central channel of the combination burner. The gas pressure upstream of the burner is 2.5-3.0 kPa. The air velocity in the narrow section of the burner is 40 m/s. The mixture (fuel oil-air or gas-air) is ignited by an electric igniter.

The gas-oil combination burner for a supercharged once-through boiler of a 300-MW monobloc unit (Fig. 8.9) has a tangential-axial air supply through a set of vanes, where the main air flow is distributed between two channels. In addition to this, ternary air is continuously supplied through the central channel to cool the fuel-oil burner. At a reduced boiler load, the air flow rate through the peripheral annular channel is diminished

by means of a control gate. The fuel oil supply is effected by means of a TKZ-4M steam-mechanical oil burner with a rated capacity of 1.28 kg/s (4.6 t/h) at a fuel oil pressure of 4.5 MPa and steam pressure of 0.2 MPa. Natural gas is introduced into the air flow from the periphery mainly through a large number of 32-mm tubes and partially through holes in the central coaxial channel.

Figure 8.10 shows a gas-oil combination burner in the once-through boiler of an 800-MW monobloc unit. The burner capacity is 5.2 t/h fuel oil. A uniform distribution of air between the burners is ensured by a common large air box for all the burners located on one side of the

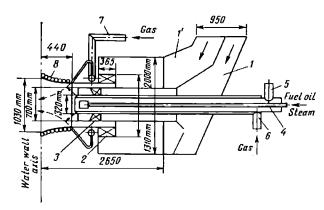


Fig. 8.9. Gas-fuel oil burner with peripheral and central gas supply
1. 1'—central and peripheral air boxes; 2—tangential vanes; 3—axial vanes; 4—channel of mechanical steam burner; 5—air inject to central air flow; 6—gas inject to coaxial channel; 7—peripheral gas supply; 8—arrangement of water wall tubes around the burner

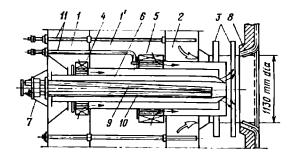


Fig. 8.10. Gas-fuel oil burner; of throughput capacity of 5.2 t/h fuel oil or 5.540 00 m³ natural gas for an 800-MW monobloc steam boiler 1. 1'—central and peripheral channels for hot air; 2—recirculating gas channel; 3—corrugated expansion joint; 4, 5—tangential whirling vanes; 6—central channel for natural gas; 7—air seal to prevent furnace gas outbreak from the burner; 8—arrangement of water wall tubes around the burner port; 9—fuel oil burner channel; 10—electric gas igniter; 11—impulse lines of air pressure control

furnace. Each air box is separated along its length into two sections to channel air into the internal and peripheral channels of burners. An additional box is provided to recirculate the flue gases to the burners. Air flows are whirled by tangential vanes, and the gases enter the furnace in a straight flow and are mixed with the expanding flow of secondary air.

Natural gas is introduced through the central coaxial channel arranged at an angle of 45° to the flow axis. The difference in the thermal expansion of the air box with burners and the furnace water walls is absorbed by means of corrugated expansion joints.

When changing to gas combustion, the oil burner is switched off automatically and retracted into the central channel. If both fuels are burned simultaneously, one of them (most often fuel oil) will not be completely burned, because of the different mixing conditions and different ignition times.

Gas-oil combination burners are designed for furnace operation with the least excess air ratio = 1.02-1.03) in fuel oil combustion, so as to minimize corrosion in the low-temperature portion of the gas path (in the region of air heater). For operation at a low excess air ratio, burners and air-supply path elements should be carefully manufactured and assembled so as to avoid a non-uniform distribution of fuel and air between the burners. All fuel nozzles are calibrated, the allowable difference in their throughput capacity being not more than 2%.



CHARACTERISTICS, PARAMETERS AND MOTION EQUATIONS OF WORKING FLUID

9.1. Principal Hydrodynamic and Heat-transfer Equations for the Water-steam Path

The hydrodynamic and heat-transfer processes in the water-steam path of a boiler take place in tubular elements. Tube walls possess such properties as heat conductivity, heat capacity and mass. The surfaces of tubes are in contact with the surroundings on the outside and with the working fluid on the inside and are thus involved in the processes occurring in these media. The properties of the tube walls and the proces-

ses occurring on their surfaces are included into the boundary conditions of the equations which describe tho processes in the water-steam path. These are the equations of material balance, of the balance of energy of the working fluid and surrounding walls, and the equation of motion (or equation of momentum).

All these equations are discussed in the courses on hydraulics and heat transfer and will not be derived here. We write these equations in their final forms:

-material balance equation:

$$\frac{\partial G}{\partial x} + f \frac{\partial \rho}{\partial x} = 0 \tag{9.1}$$

-equation of the energy balance of the working fluid flow:

$$\frac{\partial (Gi)}{\partial x} + f \frac{\partial (\rho i)}{\partial x} = q_{in} \qquad (9.2)$$

—cquation of the heat balance for channel walls:

$$q_{in} = q_{ex} - mc_m \frac{\partial t_{ni}}{\partial \tau} = \alpha \Pi (t_m - t)$$

(9.3)

In these equations, G, ρ and i are, respectively, the flow rate, density, and enthalpy of the working fluid in the channel, f is the cross-sectional area of the channel, z is the coordinate along the direction of motion of the working fluid, q_{in} and q_{ex} are the linear heat flux densities on the internal and external surface of the channel, m is the linear mass of the metal of channel walls (the mass of tube at a length of 1 m), t_m and cm are the temperature and specific heat of the tube's motal, α is the coefficient of heat transfer from the wall to working fluid, and Π is the internal perimeter of the channel cross section (for a tube, $\Pi = \pi d_{in}$).

The second law of mechanics as applied to the flow of a fluid through channels can be specifically written as follows

$$-\frac{\partial p}{\partial z} = \rho \frac{\partial w}{\partial \tau} + \rho w \frac{\partial w}{\partial z} + k \frac{w^2 \rho}{2}$$

$$\pm \rho g \sin \alpha \qquad (9.4)$$

This is the equation of motion, or the equation of momentum. Integrating equation (9.4) between the limits z = 0 ($w = w_1$, $p = p_1$)—the beginning of the channel and the end of the channel—in the general caso the section z = l ($w = w_2$, $p = p_2$) gives an integral form of the equation of motion:

$$p_{1}-p_{2} = \int_{0}^{1} k \frac{w^{2} \rho}{2} dz + \int_{w_{1}}^{w_{1}} (w \rho) dw$$
$$+ \int_{0}^{1} \rho \frac{\partial w}{\partial \tau} dz \pm g \int_{0}^{h} \rho dh \quad (9.5)$$

where $h = l \sin \alpha$.

As follows from equation (9.5), the pressure gradient in the channel, $\Delta p = p_1 - p_2$, is spent on overcoming the following forces which appear in the moving fluid:

-friction forces (the proportionality factor k is determined experimentally; it is related to the coefficient of friction by the formula $k = \lambda/d$):

$$\int_{0}^{z} k \frac{w^{2}\rho}{2} dz$$

-forces due to different velocities at the inlet and outlet of the channel, or the acceleration of the flow, i.e. the non-uniform velocity field along the channel length:

$$\int_{w}^{w} (w\rho) dw$$

-forces due to time variations of the velocity field, or the inertia component of the pressure gradient, which is other than zero in transient processes only:

$$\int_{0}^{1} \rho \frac{\partial w}{\partial \tau} dz$$

-the component of the gravity force, or hydrostatic component, which is determined by the average density and by the difference in heights, or levels (it is essentially the hydrostatic pressure of a fluid column of height h):

$$g\int\limits_{0}^{h}\rho\,dh$$

Thus, the hydrodynamic and heattransfer processes that take place in the water-steam path of a boiler can be described by a set of equations: the equation of mass balance (9.1), equation of energy balance of the working fluid flow (9.2), the equation of heat balance for the heated channel walls (9.3), and the equation of motion (9.4). They can be supplemented by the equation of state

$$\rho = \rho \ (p_{\bullet} \ i) \tag{9.6}$$

which relates a thermodynamic parameter to two other parameters, and by the empirical relationships for the heat-transfer coefficient α and the coefficient of friction, k. The set of equations (9.1) through (9.4) and (9.6) together with the empirical relationships for α and λ represent a closed system: the number of unknowns $(G, \rho, i, t_m, \text{ and } p)$ is equal to the number of equations (since $w = G/f\rho$). The external actions on the system: q_{ex} , G_{inl} , i_{inl} , and G_{out} are assumed to be known.

Under steady-state conditions, all derivatives with respect to τ in equations (9.1) to (9.4) and (9.6) are equal to zero, and all the parameters are functions of the coordinate z only, because of which the fundamental equations take the forms:

-equation of mass conservation

$$\frac{dG}{dz} = 0 \tag{9.7}$$

-equation of energy conservation

$$\frac{d(Gt)}{\partial z} = q \tag{9.8}$$

-equation of motion

$$w\rho \frac{dw}{dz} + k \frac{w^3\rho}{2} \pm \rho g \sin \alpha = -\frac{dp}{dz}$$
(9.9)

-equation of state

$$\rho = \rho (p, i) \qquad (9.10)$$

Equation (9.3) is excluded from the system (9.1)-(9.4), (9.6), since $q_{in} = q_{ex} = q$. As follows from equation (9.7), the mass flow rate under steady-state conditions is the same in all sections of the channel, i.e. G(z) = constant. Noting this, we obtain from equation (9.8):

$$\frac{dl}{dz} = \frac{q}{G} \tag{9.11}$$

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$$i = i_{InI} + \frac{q}{G}z \qquad (9.12)$$

9.2. Characteristics of Motion of a Steam-water Mixture

The principal characteristics of a steam-water flow are as follows:

The mass velocity wp of a medium (water, steam or steam-water mixture). The cooling effect of a flow depends on its density ρ , as well as on velocity w. The cooling effect of a flow is estimated in terms of the product of these parameters, i.e. through the mass velocity $w\rho$, $kg/(m^2 s)$, which is essentially the mass flow rate per unit area of the tube cross section:

$$w\rho = \frac{G}{t} \tag{9.13}$$

The circulation velocity w_0 , m/s, is the velocity of water at the saturation temperature; it corresponds to the flow rate of the working fluid in tubes:

$$w_0 = \frac{G_m}{i\rho'} \tag{9.14}$$

where G_m is the mass flow rate of the steam-water mixture through the tube system, kg/s (at the entry of water into steam-generating tubes, $G_m = G_w$), f is the free cross-sectional area for the passage of the working fluid, m^2 , and ρ' is the density of water at the saturation temperature, kg/m³.

The resolved velocity of water w'_0 , m/s, is the velocity that water would

have in a particular cross section of a steam-generating tube if it occupied that cross section completely:

$$w_0' = \frac{G_w}{to'} \tag{9.15}$$

The resolved velocity of steam w_0 , m/s, is the velocity that steam would have in a particular cross section of a steam-generating tube if it occupied that cross section completely:

$$w_{\bullet}^{\bullet} = \frac{G_{\bullet}}{t\rho^{\bullet}} \qquad (9.16)$$

where G_s is the mass flow rate of steam through the tube system, kg/s, and ρ'' is its density, kg/m³.

The relative velocity of steam w_r , m/s. The actual velocities of water and steam differ from their flow rates, since under the actual conditions of combined motion of water and steam in a mixture they move with different velocities: $w_s \neq w_w$.

In an ascending flow, steam moves faster than water, $w_* > w_w$; in a descending flow, $w_* < w_w$; their difference is equal to the relative velocity of steam:

$$w_r = w_{\bullet} - w_{\omega} \tag{9.17}$$

The relative velocity of steam has a vital effect on the motion of the steam-water mixture (see Sec. 12.1).

The mass steam content, or dryness fraction, x, is the mass fraction of the flow rate of steam in the flow rate of the steam-water mixture at $w_s = w_{to}$:

$$x = G_s/G_m \tag{9.18}$$

For a flow in thermodynamic equilibrium, the mass steam content can be expressed by the formula:

$$x = \frac{i \mu - i'}{r} \tag{9.19}$$

where i_{fl} is the unit enthalpy of the flow in a particular cross section of the tube system, kJ/kg, i' is the unit enthalpy of water at the saturation line in that section, kJ/kg, and r is the heat of the phase transition at the same pressure, kJ/kg.

The mass water content, or wetness fraction is determined by the formula:

$$1 - x = 1 - \frac{G_s}{G_m} = \frac{G_w}{G_m} \quad (9.20)$$

The velocity of steam-water mixture w_m , m/s. Using the expression for the circulation velocity, we can write:

$$w_0 = \frac{G_m}{f\rho'} = \frac{G_s + G_w^{4s}}{f\rho'}$$
 (9.21)

In turn, the mass flow rates, kg/s, of water and steam are:

$$G_w = V_w p' = w_0' \rho' f$$

 $G_\bullet = V_\bullet \rho'' = w_\bullet' \rho'' f$

where V_w and V_s are the volume flow rates of water and steam, m^3/s .

Substituting for G_w and G_s into equation (9.21), we obtain:

$$w_0 = w_0' + w_0' \frac{|\rho''|}{\rho'} \qquad (9.22)$$

The velocity of the steam-water mixture can be written as the sum of the volume flow rates of its components, $V_{\bullet} + V_{\omega}$, related to the total cross-sectional area for the passage of flow:

$$w_m = \frac{V_s + V_{\omega^{\bar{b}}}}{f_{\theta^{\sigma}}} \tag{9.23}$$

or, noting G_{ω} and G_{s} ,

$$w_m = w_0' + w_0' \qquad (9.24)$$

Substituting w'_0 from equation (9.22) into (9.24) gives:

$$w_m = w_0 + w_0^* \left(1 - \frac{\rho^*}{\rho'}\right)$$
 (9.25)

It is often essential to express w_m through the mass steam content x. Taking into account that the sum of cross-sectional areas occupied by steam and water, f_s and f_w is equal to f, we have

$$f_s w_s + f_w w_w = w_m f \qquad (9.26)$$

which may be re-written as

$$G_{\bullet}v^{\bullet}+G_{w}v^{\prime}=G_{m}v_{m}$$

whereby

$$v_m = \frac{G_s}{G_w} v'' + \frac{G_w}{G_m} v'$$

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$$v_m = xv'' + (1 - x)v'$$
 (9.27)

It is also known that

$$w_m = \frac{G_m v_m}{I} \tag{9.28}$$

Substituting v_m from equation (9.27) into (9.28), we have:

$$w_{m} = \frac{G_{m}v'}{I} \left[1 + x \left(\frac{v''}{v'} - 1 \right) \right]$$
$$= w_{0} \left[1 + x \left(\frac{v''}{v'} - 1 \right) \right]$$

0 r

$$w_m = w_0 \left[1 + x \left(\frac{\rho'}{\rho''} - 1 \right) \right] (9.29)$$

The volume steam content β. The volume fraction of steam in a steamwater mixture flow with equal velocities of water and steam is called the volume steam content:

$$\beta = \frac{V_s}{V_s + V_w} \tag{9.30}$$

Taking account of equations (9.18) and (9.20), we have:

$$\beta = \frac{1}{1 + \frac{1 - x}{x} \frac{\rho^x}{\rho'}} \quad (9.31)$$

The correlation between the mass steam content x and the volume steam content β in a steam-water mixture flow is shown in Fig. 9.1. At low pressures, the effect of x on β is especially strong at low values of x and decreases inversely with pressure. At

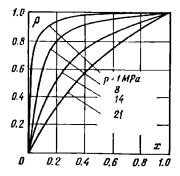


Fig. 9.1. Relation between mass steam content x and volume steam content β at various pressures

any pressure, $d\beta/dx$ decreases as x increases. These relationships have a vital effect on the regime and temperature of two-phase flows through tubes.

The actual steam content φ is the fraction of the tube cross section occupied by steam, f_a , which corresponds to the actual velocities of water and steam:

$$\varphi = f_{\bullet}/f \tag{9.32}$$

The actual steam content φ accounts for the relative velocity of steam w_r , i.e. it relates to cases when $w_s \neq w_w$. As indicated earlier, the volume steam content assumes these velocities to be equal to each other, $w_s = w_w$. The relationship between the actual steam content and volume steam content is as follows:

$$\beta = \frac{V_s}{V_s + V_w} = \frac{w_s f_s}{w_m f} = \frac{w_s}{w_m} \varphi \qquad (9.33)$$
 whereby

$$\varphi = \frac{w_m}{w_n} \beta \qquad (9.34)$$

Denoting $w_m/w_s = c$, we obtain:

$$\varphi = c\beta \qquad (9.35)$$

For an ascending flow

$$w_s > w_m$$
, $c < 1$, $\varphi < \beta$

For a descending flow

$$w_a < w_m, c > 1, \phi > \beta$$

If there is no difference in velocity, that is $w_s = w_w = w_m$, c = 1 and $\varphi = \beta$ (9.36)

The proportionality factor c characterizes the ratio of the velocity of the steam-water mixture w_m to the actual velocity of steam w_s , and thus it characterizes the relative velocity of steam w_r . With an increase in pressure, the relative velocity of steam decreases and, as pressure approaches the critical value, c tends to unity while ϕ tends to β .

The mass steam content in a uniformly heated evaporating tube varies linearly along the tube length between its extreme values: x = 0 at

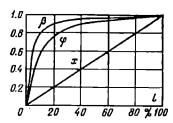


Fig. 9.2. Variations of β , ϕ and x along the length of a steam-generating channel

the inlet (if $i_{inl} = i'$) and x = 1 at the outlet (if $i_{out} = i''$), (Fig. 9.2). On the other hand, the volume steam content β and the actual steam content ϕ , which both have zero initial values, increase sharply in the initial portion of a tube. Further in the tube, they increase less intensively to the maximum value of unity at the tube end. With an increase in pressure, ϕ and β approach x.

The pressure steam content φ is an important flow characteristic, since it describes the actual distribution of water and steam in a mixture and their individual velocities in the combined flow.

The actual velocity of steam is

$$w_{\bullet} = w_{\bullet}^{\bullet}/\Phi \qquad (9.37)$$

and the actual velocity of water is

$$w_{\omega} = w_0'/(1 - \varphi)$$
 (9.38)

The apparent density of a steam-water mixture, ρ_m , kg/m³, is the density

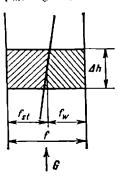


Fig. 9.3. To derivation of the formula for the density of steam-water mixture

in a flow where water and steam have the same velocity:

$$\rho_m = \frac{G_{\theta}}{w_0^* f} + \frac{G_{w}}{w_0^* f} \qquad (9.39)$$

The actual density of the steamwater mixture, ρ_a , kg/m³, i.e. the density at the actual velocities of steam and water, can be found from the following consideration. Let us separate a tube element of height Δh filled with a steam-water mixture (Fig. 9.3). Let steam and water in the element be represented by corresponding elementary volumes. The fractions of the cross sections occupied by steam and water may be denoted as f_{st} and f_{w} and their sum, as f. The masses of the two components may be written as $f_{at}\Delta h\rho''$ and $f_{\omega}\Delta h\rho'$ and their sum is the mass of the separated volume of the steam-water mixture, $\int \Delta h \rho_a$. This gives us an expression for the actual density of steam-water mixture:

$$\rho_a = \rho' - \phi \ (\rho' - \rho'') = (1 - \phi) \ \rho' \\
+ \phi \rho'' \ (9.40)$$

In an ascending tube, the steam has a relative velocity w_r , which results in a decrease in the fraction of the cross section occupied by the steam, φ , and an increase in the fraction occupied by water, $(1-\varphi)$, so that $\rho_a > \rho_m$. With increasing pressure, ρ_a tends to ρ_m .

The circulation ratio K is an inverse of the mass steam content and is essentially the ratio of the quantity of circulating water to the quantity of steam produced for the same time interval (see Sec. 1.2):

$$K = 1/x \tag{9.41}$$

9.3. Regimes of Steam-water Mixture Flow

The intensity of heat removal from a heating surface substantially depends on the conditions, or regimes, of the motion of the steam-water mixture on that surface. Under identical conditions, the structure and

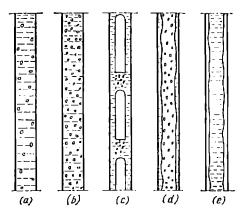


Fig. 9.4. Steam-water mixture flow modes in vertical tubes

(a) bubble flow; (b) emulsion flow; (c) slug flow; (d) dispersed annular flow; (e) inverse dispersed annular flow

regime of flow are determined by the spatial orientation of the heating tubes, which is used to organize the motion of the working fluid in steam-generating tubes.

Vertical tubes. An adiabatic flow of a steam-water mixture at a low steam content and slow velocity is essentially liquid with rare fine vapour bubbles distributed in it. This is what is called bubble flow (Fig. 9.4a). As the steam content increases and if wo is high, the moving mixture contains a larger number of fine vapour bubbles; this is the emulsion flow, or frothy flow (Fig. 9.4b). At low values of $w\rho$, an increase in steam content may result in the coalescence of fine bubbles into larger formations whose size may be comparable with the tube diameter and whose length may be many times the diameter. These formations are called 'slugs' and, correspondingly the flow regime is the slug regime (Fig. 9.4c). Behind a slug there is a thin liquid bridge containing fine vapour bubbles.

With a further increase in the steam content, water bridges between the slugs disappear and the slugs merge into a continuous vapour column with atomized water droplets in it. This

column moves along the tube core and is surrounded by a continuous annular water film which moves along the tube wall. The water film intensively cools the internal surface of the tube. This is what is called the disperse-annular (wet-wall) flow (Fig. 9.4d). The thickness of the water film depends on the ratio between the flow rates of water and steam. At a high pressure and high steam velocity, the major mass of the film is broken off and carried as droplets by the steam flow, leaving only a very thin water film on the wall, which soon evaporates. In heated channels, some specific flow regimes may take place. In film boiling, for instance, a vapour film may separate the liquid from the heat-exchange surface, while liquid fills in the core of the channel; this is called the inverse disperse-annular flow (dry-wall flow), Fig. 9.4e.

The flow regimes described above are rather conditional since they gradually change from one type to another with no distinct boundaries between them.

Horizontal tubes. The flow of the steam-water mixture in a horizontal tube is characterized by a non-uniform distribution of structural flow components over the tube cross section. Since steam has a lower density, it moves primarily along the upper tube wall, while the main mass of water is concentrated at the lower wall. The asymmetry of flow relative to the horizontal depends on the velocity of the steam-water mixture and tube diameter. At higher velocities, the flow is less asymmetrical. In steam-generating tubes 30-40 mm in diameter and at relatively high inlet velocities (w > 1 m/s), vapour bubbles form in the initial tube section. They are detached from the tube surface and move along with the liquid (Fig. 9.5a). The vapour bubbles increase in number in the direction of the flow and begin to merge into larger formations. An ever increasing quantity of steam is involved in the combined motion of

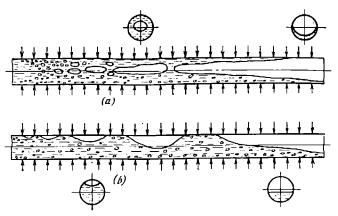


Fig. 9.5. Two-phase flow modes in horizontal tubes at (a) high and (b) low inlet velocity

the two phases, so that at a high steam content the flow in a tube becomes almost axisymmetrical and resembles the: disperse-annular regime in vertical tubes.

With a low velocity of water at the to a steam-generating tube (w < 0.5 m/s), asymmetrical motion of water and steam may result in the exposure of substantial portions of the tube surface (Fig. 9.5b). The flow then becomes essentially asymmetrical along the whole length of the tube, with the steam moving along the upper tube wall. In other words, the two-phase flow is separated, as it were, into two individual flows. This flow regime is unstable. With an increase in the flow velocity, waves may form on the separating surface with their tops periodically touching the superheated upper wall.

At supercritical pressures, the working fluid is a homogeneous medium. Nonetheless, even with a directed motion of the fluid at supercritical pressure in a horizontal channel, free convection may take place in a transverse direction, resulting in density variation along the height of the flow. This inhomogeneity can be characterized by the motion of a lighter (less dense) medium along the upper surface of a tube and of a heavier (denser) medium, along the lower surface, with no distinct boun-

dary between them. The difference in densities increases with an increase in the vertical size of a channel or of the tube diameter.

In tube bends, some portions of the tube surface may be washed less intensively by the fluid than others, which is due to the centrifugal effect by which water is thrown towards the outside surface of a bend, while the tube wall at the inside surface is insufficiently cooled by water.

9.4. Hydraulic Resistances

Since tubes offer resistance to the motion of water, steam-water mixture or steam, a pressure gradient forms between any two sections along the length of a tube. The general equation of the total pressure gradient has been given in Sec. 9.1. For practical calculations, it can be written in a more convenient form:

$$\Delta p = \Delta p_{fr} + \Delta p_I + \Delta p_{ac} \pm \Delta p_h$$
 (9.42)

i.e. the total pressure gradient between any two sections of a heated tube is the sum of pressure gradients due to friction, Δp_{fr} , local resistances Δp_{l} , acceleration Δp_{ac} , and hydrostatic head Δp_{h} .

The resistance due to friction is caused by the viscosity of the moving

fluid. For a single-phase isothermal flow in a straight channel of constant cross section, this term is found from the formula:

$$\Delta p_{fr} = \int_{0}^{l} k \frac{w^{2}\rho}{2} dz = \lambda_{0} l \frac{w^{2}}{2} \rho$$
 (9.43)

where Δp_f , is the resistance due to friction, Pa, $\lambda_0 = \lambda/d$ is the resolved friction coefficient, 1/m, l is the length of the channel, m, w is the velocity of the fluid, m/s, and ρ is the density of the fluid, kg/m^3 .

The friction resistance for a two-phase flow can be principally determined by the formula for single-phase flows, by replacing the single-phase flow velocity by the velocity of steamwater mixture w_m , provided that the two-phase flow can be considered homogeneous, i.e. obeying the following relationship:

$$\Delta p_{fr} = \lambda_0 l \frac{w_m^2}{2} \rho_m \qquad (9.44)$$

Noting the law of mass conservation (see Sec. 9.1), we can write:

$$\rho_m w_m = \rho^{\ell} w_0 \qquad (9.45)$$

Noting equation (9.25), formula (9.44) can be re-written as follows:

$$\Delta p_{fr} = \lambda_0 l \frac{w_0}{2} \rho' \left[w_0 + w_0^* \left(1 - \frac{\rho'}{\rho'} \right) \right]$$
(9.46)

or, noting equation (9.29):

$$\Delta p_{fr} = \lambda_0 l \frac{w_0^2}{2} \rho^4 \left[1 + x \left(\frac{\rho'}{\rho''} - 1 \right) \right]$$
(9.47)

In most flow regimes, the structure of a two-phase flow differs noticeably from a homogeneous structure (see Fig. 9.4), for which reason correction factor ψ is introduced into the last equation to account for the effect of the flow structure. With a constant steam content, we then have:

$$\Delta p_{fr} = \lambda_0 l \frac{w_0^2}{2} \rho' \left[1 + x \psi \left(\frac{\rho'}{\rho''} - 1 \right) \right]$$
(9.48)

The pressure loss due to friction for a flow of variable steam content can be determined from the formula:

$$\Delta p_{fr} = \lambda_0 l \frac{w_0^2}{2} \rho' \left[1 + \overline{x} \overline{\psi} \left(\frac{\rho'}{\rho''} - 1 \right) \right]$$
(9.49)

where x is the average steam content in the channel and

$$\overline{\psi} = \frac{\psi_f x_f - \psi_i x_i}{x_f - x_i}$$

where ψ_i and ψ_f are correction factors to account for the effect of the initial and final steam content, x_i and x_f , on the flow structure.

Pressure losses due to local resistances are explained by the energy consumption for detachment of the boundary layer from the tube wall and the formation of whirls in the flow. Local resistances appear in places where the shape or direction of a channel changes; conditionally, they are considered to be localized in a particular section and do not include friction resistance.

The pressure loss in local resistances to a single-phase flow is determined by a formula similar to (9.43) in which the resolved friction coefficient $\lambda_0 = \lambda/d$ is replaced by the coefficient of local resistance ξ_1 (it may be found in reference books):

$$\Delta p_l = \xi_l \frac{w^2}{2} \rho \qquad (9.50)$$

For a two-phase flow:

$$\left[\Delta p_{l} = \Sigma \xi_{l}^{\prime} \left[\frac{\omega_{0}^{2}}{2} \rho^{\prime} \left[1 + x \left(\frac{\rho^{\prime}}{\rho^{\prime}} - 1 \right) \right] \right]$$
(9.54)

where ξ_i' is a conditional coefficient of local resistance for a steam-water mixture, usually $\xi_i' > \xi_i$. For a single-phase flow across a tube bundle, the hydraulic resistance is

$$\Delta p_{tb} = \xi_{tb} \frac{w^2}{2} \rho \qquad (9.52)$$

The coefficient ξ_l for a flow across a tube bundle depends on the design of the bundle.

For a two-phase flow:

$$\Delta p_{tb} = \xi_{tb}' \frac{w_b^2}{2} \rho' \left[1 + x \left(\frac{\rho'}{\rho'} - 1 \right) \right] (9.53)$$

The pressure loss due to acceleration is caused by a change in the volume, and therefore, in the velocity of a flow. It can appear as the steam content of a flow increases due to heating, as the fluid passes through a reduced cross section, or both.

The pressure loss due to flow acceleration can be found by the formula:

$$\Delta p_{ac} = \int_{w_i}^{w_f} w \rho \, dw \qquad (9.54)$$

For a steady-state flow:

$$w_i \rho_i = w_f \rho_f = w \rho = \text{constant}$$

Therefore, at $p > p_{cr}$

$$\Delta p_{ac} = w\rho (w_f - w_l)$$

= $(w\rho)^2 (v_f - v_l)$ (9.55)

For a two-phase flow, $p < p_{cr}$. Expressing v_f and v_l through the mass steam contents:

$$v_f = v' (1 - x_f) + v''x_f$$

 $v_l = v' (1 - x_l) + v''x_l$

and substituting them into formula (9.55), we finally have:

$$\Delta p_{ac} = (w\rho)^2 (v'' - v') \times \langle x_t - x_t \rangle \quad (9.56)$$

The hydrostatic component of pressure loss for single- and two-phaso vertical flows is:

$$\Delta p_h = g \int_0^h \rho \, dh = \bar{\rho} \, gh \quad (9.57)$$

For a single-phase flow, ρ is taken as the average density of the fluid in section h, and for two-phase flows is determined by formula (9.40).

When determining the total pressure gradient, Δp_h is assumed to be positive for ascending flows since it increases the total resistance of a tube, and negative for descending flows where it ceases to be a resistance and, on the contrary, acts in the direction of the flow.

9.5. Thermophysical Properties of Working Fluid in the Path of a Monobloc Unit

Variations of the parameters and physical properties of the working fluid in various sections of the watersteam path of a supercritical-pressure monobloc unit are shown in Fig. 9.6. The highest pressure of the fluid is at the inlet to the high-pressure water heater, downstream of the feed water pump, p_{ioh} , and the lowest pressure is in the turbine condenser, pc. For supercritical-pressure plants, this pressure range is from 32 MPa to 0.003 MPa and for high-pressure power plants, from 17 MPa to 0.003 MPa. In the condensate path, which includes a condensate pump, and in the feed water path with a feed water pump, the

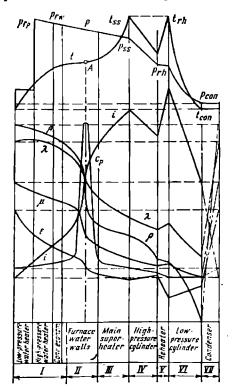


Fig. 9.6. Variations of parameters and thermophysical properties of the working fluid in the water-steam path of a supercriticalpressure monobloc unit

pressure of the turbino condensate is raised to a value equal to the pressure at the inlet to the high-pressure water heater, which ensures the motion of the working fluid through the entire water-steam path of the boiler.

In the turbine, steam moves due to a pressure gradient between the inlet, where the pressure is equal to that downstream of the boiler, and the outlet, where it is equal to the pressure in the condenser.

The highest temperature of the working fluid is the temperature of superheated steam and the lowest is that in the condenser. In high-pressure and supercritical-pressure plants, this temperature interval is from 545° to 30°C.

Turbine condensate is preheated in the low-pressure water heaters and the deacrator of the condensate path from 30° to 105-165°C and further in the high-pressure water heaters of the feed water path to the feed water temperature t_{fw} (usually 145-270°C). Water pressure in this path varies from p=0.3-1.0 MPa to 15-17 MPa in high-pressure plants or to 30-32 MPa in supercritical-pressure plants.

The total pressure gradient in the path, $\Delta p = \Delta p_{ec} + \Delta p_{dea} + \Delta p_{sh}$ constitutes 20-30% of the pressure of superheated steam. On the other hand, the temperature of the working fluid is raised substantially, from t_{fw} to superheated-steam temperature t_{ss} .

Superheated steam is fed into the turbine with the parameters t_{aa} and p_{aa} . As steam expands adiabatically in the turbine, its pressure and temperature decrease. In power plants of high and supercritical parameters, the total efficiency is increased by taking off steam from an intermediate stage of the turbine (at a pressure of 3-4 MPa and temperature of 290-320°C) and recirculating it back into the boiler where it is reheated, usually to the initial steam temperature, 545°C, in an intermediate steam superheater (reheater).

Since the working fluid changes its pressure and temperature in the water-steam path of a supercritical monobloc power unit, this involves certain changes in its thermophysical properties. As shown in Fig. 9.6, the water-steam path can be divided into seven zones in which typical changes in the thermophysical properties of the working fluid take place.

The first zone includes the condensate and feed water path of the power unit with convective and radiation-heated economizers. In this zone, feed water pumps create the highest pressure in the water-steam path needed to produce steam of the required pressure. The working fluid in this zone remains liquid and therefore, its thermophysical characteristics are changed relatively weakly, notwithstanding the high rise in pressure; its dielectric permittivity ε , however, decreases to a small fraction of its initial value.

The most typical is the second (near critical) zone which is usually located in the boiler furnace. In this zone, the change from water to steam takes place, so that even a small rise in the temperature of the flow causes sharp changes in the thermophysical characteristics: near the temperature of the highest heat capacity, the enthalpy of the flow increases sharply, while the viscosity μ , heat conductivity λ , and, what is especially important, the density ρ and dielectric permittivity ϵ decrease substantially, though smoothly, not jumpwise.

In the third zone, which includes the main superheater, the temperature and enthalpy of the superheated steam increase further, but all other parameters, including density and dielectric permittivity, change less than in the previous zones.

In the fourth zone (high-pressure cylinder of turbine), the thermal energy of the superheated steam is converted into mechanical work and its pressure and temperature decrease. Accordingly, the enthalpy and density of the steam decrease, the dielectric permittivity increases somewhat (due to temperature reduction), and the

other properties (heat conductivity, heat capacity, and dynamic viscosity) continue to decrease smoothly.

In the fifth zone (roheater), steam temperature is raised again at an almost constant pressure roughly 3.5 MPa, resulting in an appreciable increase in its enthalpy and slight increase in viscosity and heat conductivity. The dielectric permittivity of steam decreases by 67-80%, whereas the density and isobaric heat capacity diminish only slightly.

In the sixth zone (low-pressure cylinder of turbine), the physical parameters of steam are changed in a manner similar to the fourth zone, the only difference being that the pressure changes much more substantially than in the fourth zone. For that reason, the density of steam decreases roughly to 1/300 or 1/400 of its initial value. Its enthalpy also decreases substantially, while changes in the other properties are rather smooth.

The seventh zone includes the turbine condenser. The condensing steam here has constant parameters: pressure 0.003 MPa and a temperature of around 30°C. Only the enthalpy and steam content of the flow are diminished. As steam is condensed fully, the density, heat conductivity, heat capacity, viscosity and dielectric permittivity increase sharply.

The specifics of subcritical pressure units are most pronounced in the second zone of the water-steam path, which is within the boiler furnace (Fig. 9.7). This zone is characterized by the conversion of water to steam (boiling); steam generation takes place at a constant temperature t' with two phases: water and steam, existing simultaneously (curve 1). For comparison, the figure shows temperature variations in the working fluid at supercritical pressure (curve 2).

As may be seen from Figs. 9.6 and 9.7, the thermophysical characteristics of the working fluid in the zone of high heat capacities at supercritical pressures vary smoothly (though rapidly), whereas at subcritical pres-

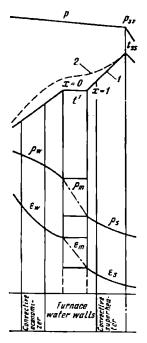


Fig. 9.7. Variations of density ρ and dielectric permittivity ε in evaporating zone 1—temperature of the working fluid at subcritical pressure; β—ditto, at supercritical pressure (for comparison)

sures the change from the properties of water to those of steam suddenly takes place at the saturation temperature. As seen in Fig. 9.7, as the water to steam transition takes place, the density of the water, ρ_w , abruptly changes to that of steam, p, which results in an increase in the enthalpy of the working fluid. Similarly, the dielectric permittivity of water changes abruptly to that of steam. In the zone of constant temperature at the saturation line and constant pressure, both phases (water and stoam) co-exist in the whole range of steam contents of the two-phase flow, 0 < < x < 1. The dot-and-dash lines in the figure show variations in the density of the steam-water mixture, ρ_m , and in its dielectric permittivity ε_m along the flow.

The thermophysical properties of water and steam are determined by

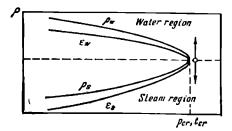


Fig. 9.8. Effect of pressure on the density of water and steam, ρ , and dielectric permittivity ϵ at the saturation line

pressure. Figure 9.8 shows variations in the density and dielectric permitti-

vity of water and steam; as may be seen, these characteristics are substantially different in a wide range of subcritical pressures, but come closer and closer together in the critical point.

The above relationships between the parameters and properties of the working fluid determine the hydrodynamic and heat-transfer processes on heating surfaces and the carry-off by steam of impurities present in water; they also play a vital part in the formation of deposits on boiler and turbine elements.



TEMPERATURE CONDITIONS ON HEATING SURFACES

10.1. Classification of Heating and Cooling Modes

A knowledge of temperature distribution in the metal of heating surfaces, which operate under complicated and heavy conditions, is essential for estimating the reliability of a steam boiler.

In boilers operating on organic fuel, there are three different regions of heat transfer. The first region includes the heating surfaces (water walls) arranged in the furnace; they receive heat mainly by radiation. The most important factor of heat transfer in this region is the pattern of distribution of the heat flow along the height of the furnace (see Fig. 20.2) and over the cross-sectional periphery of waterwall tubes (see Sec. 10.4).

The second heat-transfer region comprises the semiradiant heating surfaces which are arranged in the boiler zone where radiation from gaseous volumes is still substantial and the gas temperature is quite high (1 200-800°C). This group of heating surfaces includes primarily platens and water walls in the horizontal duct of the furnace.

The heating surfaces in the third region receive heat mostly by convection. This region has a relatively low temperature of combustion products (800-900°C at the inlet to the convective shaft and up to 100-150°C at its outlet) and accommodates convective heating surfaces: an economizer, air heater, and some superheater banks. These heating surfaces have a small tube pitch, i.e. small spaces between tubes for the passage of hot gases. Under such conditions, the fraction of radiant heat transfer from combustion products onto tubes is not high.

In the steam generators of nuclear power stations, two regions of heat transfer are usually distinguished. The first region includes the heating surfaces arranged in the reactor core. The heat in the reactor is liberated due to nuclear fission in a very restricted volume around the place where nuclear fission occurs. This makes it possible to assume that practically all the heat released by the core is concentrated in the volume of fuel elements.

The maximum amount of heat released, q_{\max} , through the surface of fuel elements is in mid-height and the zero value (in the absence of reflectors), at the end faces. Reflectors can substantially equalize the intensity of heat released.

In the second region, heat transfer takes place by convection at a moderate temperature of the heat-transfer agent (water, liquid metal or gas). For instance, in a water-heated primary circuit, water temperature does not exceed 330-350°C (at a pressure of 13-15 MPa); in circuits heated by liquid metal or gas, this temperature may be higher, up to 700-800°C. Under such conditions, there is no immediate danger of tube burn-out should the heat-transfer conditions worsen, but the durability of the heating surfaces' metal can be impaired and cause emergency situations.

Present-day engineering has made possible the release of enormous quantities of heat in the boiler furnace or the core of a nuclear power reactor. Irrespective of the type of power plant, the heat released must be actively removed through heating surfaces. This is ensured by the motion of the working fluid at a definite velocity.

The motion of the steam-water mixture, and therefore, cooling of the evaporating tubes of steam boilers and steam generators can be organized in a different manner (Fig. 10.1). In free-circulation plants, the steamwater mixture moves under the free-circulation head that appears when the tubes heat up. With an increase in load, the mass flow rate at the inlet to the evaporating tubes first

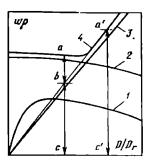


Fig. 10.1. Effect of boiler load on mass velocity

I—gravity circulation; I—multiple forced circulation; J—circulation in once-through boiler;
4—combined circulation

increases sharply to a maximum value and is then almost stabilized or even decreases somewhat, since the increasing steam generation at a high unit volume of steam results in an increased resistance of the tubes (curve *I* in Fig. 10.1).

In multiple forced circulation plants, the circulation of the water and steam-water mixture in evaporating tubes is effected by a circulation pump, so that the mass flow rate of the working fluid is almost constant irrespective of boiler load (curve 2).

In once-through boiler plants, the mass flow rate is proportional to the boiler load (curve 3). At a low load, the mass flow rate may turn out to be inadmissibly low and cause burnout of tubes. For this reason, load shedding is restricted to no less than 30% of the rated load.

Curve 4 in Fig. 10.1 depicts the characteristic of a boiler plant in which the motion of the working fluid is based on the principle of multiple forced circulation at low loads and on the once-through principle at high loads. When operating on the principle of multiple forced circulation, the mass flow rate is expressed by the sum of the ordinates ab—the mass flow rate corresponding to recirculation through the steam-generating surfaces, and bc—the mass flow rate in the economizer and superheater in once-through operation. With a chan-

ge to the once-through operation, the mass flow rate is the same in all the heating surfaces of the water-steam path, the ordinate a'c'. Thus, the combined circulation system ensures adequate cooling of all the heating surfaces irrespective of the load. Accordingly, forced-circulation single-pass motion of water and steam takes place in the economizers and superheaters.

The thermal conditions in the tubes of the steam-water path of a steam boiler or steam generator are established depending on the ratio between the quantity of heat supplied by the heat-transfer agent to the outside surface and the quantity of heat removed by the working fluid from the inside surface. The simultaneous processes of heat release and heat removal ensure that any point of a heating surface is in the necessary thermal state and has a specific temperature. The temperatures in various points of a heating surface form what is called the temperature field.

One can distinguish between a temperature field along the motion of the working fluid, i.e. along the length of tubes (assuming that they are heated uniformly around their periphery) and a temperature field across the cross section of a tube (noting that heat supply or heat removal may be uneven around the periphery of a tube).

10.2. Heat-transfer Crisis in Evaporating Tubes

For a given heat release intensity, the temperature conditions on an evaporating tube are mainly determined by the steam content, mass flow rate and pressure of the flow.

Various inadequate temperature conditions which may appear in ovaporating tubes can result in a substantial rise in the temperature of the metallic tube wall which can sometimes cause tube burn-out. One should differentiate between the inadequate temperature conditions in bubble flow and in disperse-annular flow.

With bubble flow of a steam-water mixture having a moderate steam content, a thick layer of liquid covers the tube wall (Fig. 10.2a) and the temperature of the tube is maintained at a substantially low level (section 1-2). Even with intensive heating, it exceeds the saturation temperature only by a few tens of degrees.

As the working fluid moves and is heated in the tube, more and more

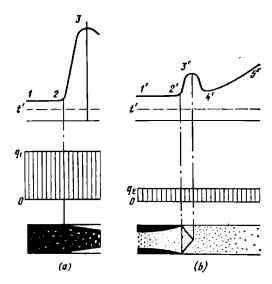


Fig. 10.2. General pattern of temperatures in a steam-generating tube at (a) bubble flow and (b) dispersed annular flow of steam-water mixture

vapour bubbles appear on the tube walls; they coalesce with one another and may eventually form a continuous vapour film which will separate the flowing liquid from the heated tube wall and thus sharply impair the conditions of heat transfer. The effect of a sharp worsening of heat transfer on a change from bubble boiling to film boiling is called the boiling crisis (burn-out conditions) and the corresponding heating load is called the critical load q_{cr} .

On occurrence of the boiling crisis (point 2), the layer of superheated steam at the heated wall becomes thicker, the heat-transfer coefficient drops down sharply, resulting in a sharp rise in the wall temperature and often in burn-out of the tube wall (point 3). These temperature conditions can occur in bubble flow if the heating rate of one of the evaporating tubes is too high and a thick layer of steam accumulates on the superheated tube wall.

In disperse-annular (wet-wall) flow, saturated steam with suspended liquid droplets moves in the core of a tube and a water film flows along the tube walls (Fig. 10.2b). Without heating, the thickness of the liquid film depends on the ratio between the flow rates of water and steam, the quantity of moisture that is separated from the flow core and wets the wall, and the quantity of moisture removed from the wall due to break-off and mechanical carry-off by the steam flow. In heated tubes, the intensity of heating has a strong additional effect on the thickness of the water film. A continuous water film can still ensure proper heat removal from the heated surface and the wall temporature can be maintained at an allowable level (line 1'-2'). In further heating, the water film becomes thinner owing to evaporation and break-off, so that only a very thin film (micro-film) remains on the wall. Under a particular heating load, the microfilm is destroyed and separated into individual islets and streaks whose number and size diminish in the course of further vaporization. Thus, a continuous steam layer forms at the wall, while the core of the flow contains a slightly superheated steam with water droplets whose concentration decreases gradually in the direction of the flow. Under such conditions, water droplets do not reach the tube walls. Besides, on entering a hotter zone of the tube they evaporate. Since vaporization is more intensive at the walls, the water droplets are repulsed from tube walls back into the steam flow. Under such conditions, heat transfer occurs through a continuous steam layer, i.e. the heat-transfer coefficient decreases sharply and the wall temperature increases (point 2' in Fig. 10.2b), resulting in a boiling crisis. In contrast to the previously mentioned case of a boiling crisis at a moderate steam content during a change from bubble to film boiling, this type of boiling crisis is caused by a change from disperse-annular to disperse flow with a complete drying out of the liquid film at a high steam content of the flow. This type of boiling crisis can even take place at a low heating load and relatively high heattransfer coefficient as the liquid phase evaporates almost completely. Since the unit volume of steam is much higher than that of water, the linear velocity of the flow increases substantially. In a boiling crisis when the liquid film dries out, the temperature of the tube wall rises less significantly (line 2'-3'), as in the former case.

Boiling crisis in the one-sided heating of evaporating tubes. We have discussed the mechanism of boiling crisis in an ascending flow of working fluid in vertical tubes heated uniformly all over their periphery.

In modern powerful steam boilers, the main steam-generating surfaces are formed by furnace water walls which are unevenly heated over their periphery. The distribution pattern of the heating load over the periphery of a water-wall tube is shown in

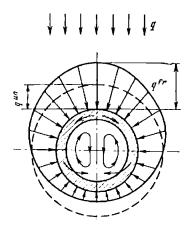


Fig. 10.3. Heat distribution in a smooth tube heated from one side

Fig. 10.3. The front side of the tube, which faces the furnace, is heated most intensively and the rear side, least intensively. Uneven heating causes transverse circulation of the working fluid from the less heated to the more heated portions of the tube periphery; some heat is also transferred by conduction in the metal wall. These two circumstances enhance the cooling effect of the working fluid, thus retarding the appearance of a boiling crisis. For this reason, burn-out conditions in tubes that are unevenly heated across their periphery can appear at substantially higher values of qcr than in those which are heated uniformly from all sides. This means that one-sided heating surfaces can be designed for more forced heating, $q_{cr}^{fr} > q_{cr}^{un}$ (see Fig. 10.3). The allowable heating load to avoid a boiling crisis can also be increased by artificial turbulization of the flow with helical inserts, screw threading of the internal surface of tubes, etc.

10.3. Temperature Conditions Along the Length of a Channel

Straight channels. Curves of temperatures in the water-steam path of subcritical-pressure drum-type and once-through boilers are shown Fig. 10.4. In boilers of any type, the temperature of feed water in the economizer, t_{fw} , is raised to the saturation point (in the limit), so that the heat absorbed by evaporating tubes is almost fully spent on vaporization, and the temperature of the steam-water mixture t' remains almost constant. In the superheater, the temperature of steam is further raised to the rated value t_{ss} .

In the economizer section I, the temperature of the flow t_i and that of the tube wall, t_i , are below the saturation temperature: $t_i < t'$ and $t_t < t'$. The thermophysical properties of water in the layer near the wall vary only slightly with temperature. Under such conditions, heat transfer is mainly governed by the laws of convection:

$$Nu_f = cRe_f^{0.6}Pr_f^{0.4}$$
 (10.1),

Where

$$\mathrm{Nu}_{f} = \frac{\alpha d}{\lambda_{f}}$$
, $\mathrm{Re}_{f} = \frac{w \rho_{f} d}{\mu_{f}}$, $\mathrm{Pr}_{f} = \frac{\epsilon_{pf} \mu_{f}}{\lambda_{f}}$

The subscript 'f' implies that the flow temperature is taken as the decisive parameter. For straight channels, C = 0.023.

Heat exchange occurs at a temperature gradient $\Delta t = t_1 - t_1 = q/\alpha_2$

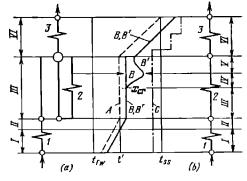


Fig. 10.4. Temperature variations in the water-steam path of (a) drum-type and (b)once-through subcritical-pressure boilers 1—economizer; 2—evaporating tubes; J—superheater; A—temperature of the working fluid; B—wall temperature of a drum-type boller; B'—wall temperature of a once-through boiler; C—allowable metal temperature

In the economizer section, heat exchange between the wall and the water takes place at a low heating intensity $(q \approx 5\text{-}10 \text{ kW/m}^2)$ and high values of the heat-transfer coefficient $(\alpha_2 = 2\text{-}5 \text{ kW/(m}^2 \text{ K}))$, so that the wall temperature is only a few degrees higher than that of the water.

In section II, the temperature of the flow t_f is lower than t', while the wall temperature is higher than t'. This means that vaporization (surface boiling) has already started on the wall. As steam bubbles are formed, they pass into the water flow and are condensed. In this section, the water flow is heated in the temperature range up to t'. This section too, belongs to the economizer, but the heat transfer in it is more intensive than in section I.

As heat is further supplied, stable bubble boiling begins in the tube: $t_f = t'$ and $t_t > t'$. In this boiling zone (section III), heat transfer takes place irrespective of the flow velocity (at velocities typical for steam boilers), but is determined by the heating intensity q and the thermophysical properties of the liquid and steam at the saturation line, which depend uniquely on pressure p. In the pressure range 0.4-16 MPa, the heat-transfer coefficient in the zone of intensive boiling can be approximated by the empirical formula

$$\alpha_{boil} = 0.34 p^{0.43} q^{0.7} \qquad (10.2)$$

Heat transfer in this section takes place at a substantially high heattransfer coefficient $[\alpha_2 = 50-100]$ kW/(m² K)], and therefore, the temperature of the tube wall only slightly exceeds the temperature of the flow even with very intensive heating, as is possible in high-forced furnaces of steam boilers without running the risk of scale deposition (see Sec. 10.4). Such conditions exist along the whole length of evaporating tubes in drumtype boilers (section III) where the mass steam content x at the outlet is rather low (less than 20%) due to the high circulation ratio. In oncethrough boilers, the steam content varies along the length of the evaporating tubes in the range 0 < x < 1. In tube portions where x is relatively low due to extensive boiling, α_z is rather high and can be calculated by formula (10.2). Under such conditions, t_t is close to t'.

Beginning from a certain steam content x_{cr} , which depends on pressure and heating intensity, the temperature of the wall rises substantially (boiling crisis associated with water film drying, see Sec. 10.2), which is indicative of deterioration of the heat exchange conditions (section IV). For this section, i.e. in the zone x > $> x_{cr}$, or upon occurrence of a boiling crisis, it may be assumed with a good approximation that the coefficient of heat transfer from the wall to steamwater mixture (α_f) varies roughly proportionally to the linear velocity of the steam-water mixture w_m to the power of 0.8; a similar relationship may be assumed from the coefficient of heat transfer to dry saturated steam α'' . Therefore:

$$\frac{\alpha_f}{\alpha''} \approx \left(\frac{w_m}{w''}\right)^{0.8} \tag{10.3}$$

Assuming that the steam-water mixture in the zone of deterioration of heat transfer moves as a homogeneous medium, the coefficient of heat transfer can be determined as for dry saturated steam whose average velocity is equal to the velocity of the mixture:

$$\frac{w_m}{w''} = x + \frac{\rho''}{\rho'} (1-x)$$
 (10.4)

The formula for α_f in the zones of deterioration of heat transfer has been proposed by Z. L. Miropolsky [61]. It includes a correction factor y to account for inhomogeneity of the fluid in the zone at a steam content that is slightly less than unity:

$$Nu_{f} = c (Re'')^{0.8} Pr_{t}^{0.8} \times \left[x + \frac{\rho''}{\rho'} (1 - x) \right]^{0.8} y (10.5)$$

where

$$Nu_f = \frac{\alpha_f d}{\lambda^*}$$
 and $Re'' = \frac{(w\rho) d}{u''}$

The correction factor y, which depends on the steam content and ρ'/ρ'' ratio, can be found by the empirical formula:

$$y = 1 - 0.1 \left(\frac{\rho'}{\rho''} - 1\right)^{0.4} (1 - x)^{0.4}$$

On attaining a maximum, the temperature of the heated tubes decreases. This is due to the fact that heat transfer is more intensive with an increase in the linear velocity of steam in the zone where the liquid phase is evaporated completely (section V). At the boundary between sections III and IV for drum-type boilers or V and VI for once-through boilers, the mass steam content becomes equal to unity (x = 1) and the enthalpy is i'', so that any further supply of heat results in steam superheating.

In all types of boilers, in the suporheater region (section VI), heat transfer to superheated steam again worsens and the temperature of the tube metal increases. In this region, the laws of convective heat exchange for a single-phase medium become valid again—but for superheated steam we have:

$$Nu_s = cRo_s^{0.6}Pr_l^{0.8}$$
 (10.6)

Under impairment of heat transfer conditions, the coefficient α_1 is substantially lower than in intensive boiling, but is still sufficiently high to ensure reliable operation of the metal of the heating surfaces at a properly selected mass velocity of the flow (see Scc. 10.2).

With an increase in pressure, surface tension decreases, so that impairment of heat transfer occurs at a lower value of x_{cr} . The same effect on x_{cr} has an increase in the heating load (due to quicker evaporation of moisture in the tubes).

The boundary of the transition to deteriorated heat-transfer conditions can move along the length of an eva-

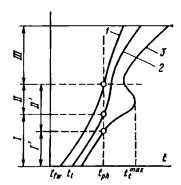


Fig. 10.5. Wall and flow temperatures in the water-steam path of a supercriticalpressure boiler

1—flow temperature; 2—wall temperature at low q; j—wall temperature at high q

porating tube depending on certain operating factors. The metal at that boundary is subjected to temperature changes which may cause thermal fatigue. The amplitude of temperature variations can be decreased by limiting the temperature gradient between the internal wall of a tube and the flow in the transition region ($\Delta t \leq 80^{\circ}$ C). This is attained by maintaining a sufficiently high flow velocity.

Curves of the temperature variations of the flow and tube wall in a vertical once-through circuit of supercritical pressure at various heating intensities are shown in Fig. 10.5. The entire region of steam generation car be divided into three heat-transfer regions: I or I'-water preheating; II or II'—pseudo-boiling; and III steam superheating. In region I, the temporature of the flow and wall at a given pressure is below the temperature of conditional phase transition, t_{ph} . The thermophysical properties of the working fluid in the boundary layer on the wall vary only slightly. Under such conditions, heat transfer is governed by convection and can be calculated by formula (10.1). The wall temperature in region I increases slowly, following variations in the flow temperature.

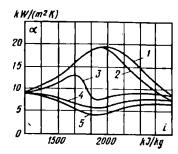


Fig. 10.6. Effect of the working fluid enthalpy and of heat flux on the heat-transfer coefficient at supercritical pressure p = 24 MPa, $\omega_{\rm P} = 700$ kg(m³ s); q, kW/m³: I = 350; z = 520; z = 640; d = 750; z = 870

In region II, the temperature of the flow remains lower than t_{ph} , but the temperature of the wall is higher than t_{ph} . For this reason, the thermophysical properties in the boundary layer and in the main flow may differ substantially (see Sec. 9.5) resulting in the features specific to the heat transfer in that region: heat transfer is intensified at low values of q (line 2 in Fig. 10.5) and, on the contrary, is impaired at high values of q (line 3). Figure 10.6 shows the effect of q on α_2 in the phase transition region. As may be seen, a2 has a maximum at a low q and decreases to a minimum at higher values of q. With a decrease in the flow velocity or an increase in the heating load, the region of unit enthalpies typical of the impaired heat transfer becomes larger.

At supercritical pressures, heat transfer in this region is caused by two main factors: the effect of varying properties of the flow on the processes of turbulent mass exchange and the appearance of noticeable thermogravitational forces (free convection). Thermogravitational forces can be characterized by the ratio Gr/Re². At Gr/Re² < 10⁻², the effect of free convection can be neglected and heat transfer in a vertical ascending flow can be calculated by the formula proposed by V.S. Protopopov:

$$Nu_s = Nu_0 \left(\frac{\bar{c}_P}{c_{Pf}}\right)^n \left(\frac{\rho_f}{\rho_f}\right)^m$$
 (10.7)

where Nu₀ is the Nusselt number under isothermal conditions:

$$Nu_0 = \frac{\frac{\xi}{8} \operatorname{Re}_{f} \operatorname{Pr}_{f}}{1.07 + 12.7 \sqrt{\frac{\xi}{8} (\operatorname{Pr}^{2/3} - 1)}}$$
(10.8)

Other quantities needed for the calculation: Gr is the Grashof number:

$$Gr = \frac{g(\rho_f - \rho_t) d^3}{v_f^2 \rho_f}$$

 ξ is the coefficient of hydraulic resistance, $\xi = (1.82 \log \text{Re} - 1.64)^{-2}$; c_p is the mean specific heat:

$$\bar{c}_p = \frac{t_l - t_l}{t_l - t_l}$$

 c_{pf} is the specific heat of the flow; and ρ_f and ρ_t are the flow density at the flow temperature and wall temperature, respectively. The exponents n and m can be found by empirical formulae.

At Gr/Re² = 10⁻²-0.4, the contribution of gravitational forces is large and heat transfer deteriorates owing to a partial degeneration of the turbulence in an ascending flow. For such conditions, the calculation should be corrected for the effect of free convection, i.e. for the ratio Gr/Re².

The conditions described by curve 3. (see Fig. 10.5) can appear at $q > q_b$ at a given value $w\rho$ or at $q/w\rho > 1q (w\rho)l_b$ [22].

In region III, the temperature of the flow exceeds t_{ph} and heat transfer is governed by the laws for superheated steam [formula (10.6)].

Bent channels. The tubing of a boiler or steam generator has straight tubular portions and numerous tube bends, such as bent tubes in cylindrical cyclone primary furnaces, coiled tube banks in nuclear steam generators, bends in multi-pass sections of water wall, bent tubes around burner ports, manholes, etc.

The motion of fluids in bent channels can be characterized by the appearance of centrifugal forces directed towards the outside of a bend. The

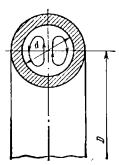


Fig. 10.7. The working fluid circulation across a curvilinear channel

fluid in the core of a flow, which is denser and therefore has a higher kinetic energy, is thrown by these forces to the outside of a bend where it displaces the fluid of a lower density. The displaced fluid moves in symmetrical currents along the side walls of the channel towards the inside of the bend. The fluid that moves from the flow core loses part of its energy in friction on the wall and is continuously displaced by new portions which move from the core. Therefore, in addition to the main axial flow, the transverse motion of two opposing symmetrical vortices (secondary circulation) appears in bent channels (Fig. 10.7).

As the two opposing vortices circulate, there remains a narrow band between them on the inside wall which cools less intensively than that would otherwise, and, as a result, heat transfer in this local zone noticeably deteriorates.

Under identical conditions, a bent channel has a higher hydraulic resistance than a straight tube. This can be explained by the fact that there is a stronger energy exchange across the flow due to the transverse circulation of the fluid.

The critical Reynolds number of the passage from laminar to turbulent flow is higher for bent channels than for straight tubes, the difference being more noticeable at a higher d/D ratio, i.e. greater curvature.

On the whole, secondary circulation intensifies heat transfer. On the other hand, it redistributes the velocities and densities of fluid in the channel cross section in such a manner that heat transfer becomes non-uniform around the channel periphery. The conditions for heat transfer, and, thus, the temperature conditions deteriorate on the inside of a channel; on the outside, heat transfer conditions are better and the wall temperature is lower.

10.4. Temperature Conditions Around the Periphery of a Channel

Figure 10.8 shows the fields of absorbed heat on the outside surface of a smooth tube and a finned tube with symmetrical fins. The tubes are heated from one side in the water wall of a boiler furnace. As may be seen, though the distribution of heat flows over the periphery of the two tubes is different, the heating load near the front point is essentially the same in both. The difference in the heating load over the periphery of both tubes results in the appearance of a heat flow through the metal from the front zone,

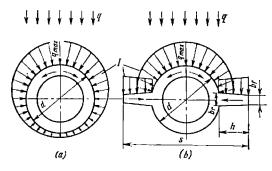


Fig. 10.8. Thermal fields in water wall tubes

(a) seventh tubes (b) finned tubes I—diag-

(a) smooth tube; (b) finned tube; I—diagram of supplied heat

where the heating load is the highest, to the rear portion of a tube, where the heating load is substantially lower. In the finned tube heat also flows along the fins from the top to the base (root). As a result, the heating load at the fin root substantially increases and under particular conditions the temperature in that zone may turn out to be even higher than on the front wall.

For reliable operation of the tube metal, heat removal from the heating surface should occur at a tolerable temperature, i.e. a temperature that is admissible for long-term normal operation of the metal. For this purpose, the concept of the design temperature of tube metal, t_t , is introduced, which is understood to be the highest local temperature as determined by considering the non-uniform heat absorption over the cross section of a gas duct or the periphery of a tube, heat flow through the wall, and hydraulic and structural non-identity of the tube coils.

Smooth tubes. When the thermal field of a tube is uniform, the wall temperature on the inside depends on the temperature of the working fluid t_{fl} , heating intensity q (on the inside surface, q_{in}), the coefficient of heat transfer from the inner wall to the working medium α_2 and, if there are no internal deposits, can be found from the formula:

$$t_{i}^{in} = t_{fl} + \frac{q_{in}}{a_{2}}$$
 (10.9)

To calculate the long-term strength of tubes, the average temperature of the metal across the tube thickness is determined

$$t_t = t_{fl} + q \left[\frac{1}{\alpha_2} + \frac{\delta_t}{(1+\beta)\lambda_t} \right] \quad (10.10)$$

The temperature on the outside surface of a tube is higher than the average temperature. It is used for estimating the probability of scaling. For a tube that is heated uniformly all over its periphery, noting the thermal resistance of the tube wall,

this temperature can be found from the formula:

$$t_t^{out} = t_{fl} + q \left[\frac{1}{\alpha_2} + \frac{2\delta_t}{(1+\beta)\lambda_t} \right]$$

$$(10.11)$$

Apart from the quantities already known, here δ_t and λ_t are the thickness and heat conductivity of the tube wall, and $\beta = d_{out}/d_{in}$ is the ratio of the outside diameter to the inside diameter of the tube.

For operation of the metal at a given α_2 , the combination of a high temperature of the working fluid and a high heating intensity is most unfavourable. For this reason, the reliability of metal operation in heating surfaces is estimated by checking the wall temperature at various combinations of t_{fl} and q that might occur in a particular boiler element.

The temperature field of a tube may be non-uniform over the periphery of its cross section if heat is supplied non-uniformly to the external surface (one-sided heating of furnace water walls, cross-flow around tubes of a convective heating surface) or if heat is removed non-uniformly from the internal surface (separation of phases at subcritical pressures or density separation at supercritical pressures). A non-uniform temperature field in the cross section of a channel results in heat spreading through the wall from regions at the highest temperature to those where the metal temperature is lower. Noting the effect of heat spread, formula (10.11) takes on a new form:

$$t_t^{out} = t_{ft} + \mu \beta q_{\max} \left[\frac{1}{\alpha_2} + \frac{2\delta_t}{(1+\beta)\lambda_t} \right]$$
(10.12)

where q_{max} is the heating load at the point of maximum heat release, and μ is the coefficient of heat spread over the tube cross section.

The coefficient of heat spread μ for a particular tube point is understood as the ratio of the actual temperature to the temperature that would appear at that point should the heating be uniform with a heat flow q_{max} .

Non-uniformity of the temperature field is most essential for horizontal tubes of subcritical-pressure plants and also for tubes of any orientation of supercritical plants in zones where the working fluid has a high heat capacity.

At supercritical pressures, the thermophysical properties of the working fluid may substantially change in a rather narrow temperature range (Fig. 9.6). In vertical heated tubes, this may result in a density gradient along the radius and the formation of a zone of lower density and lower conductivity near the heating surface whose temperature is naturally higher. This can lower the intensity of heat transfer and worsen the temperature conditions on the heating surface.

In horizontal tubes of a diameter more than 15-20 mm, even if heated uniformly over the periphery, internal heat transfer in the upper portion is much worse than in the lower, which is due to asymmetry of the mixture flow under the effect of gravitational forces. For this reason, the range of deteriorated heat transfer is larger in horizontal tubes than in vertical. In inclined tubes, even when heated uniformly over the periphery, unsymmetrical two-phase flow can also appear under certain conditions, resulting in poor heat transfer in the top portion of a tube. Thus, the region of deteriorated heat transfer is larger in inclined tubes than in vertical, but smaller than in horizontal tubes.

At subcritical pressures and separated flow of the working fluid, heat transfer is unsymmetrical and the temperatures on the upper and lower surfaces of tubes are different (Fig. 10.9). Heat transfer at the upper tube surface, where the wall temperature is close to the saturation point, is worse than at the lower surface. This is associated with particular conditions of two-phase flow in horizontal tubes where the flow can be separated

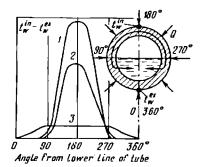


Fig. 10.9. Overheating of the upper surface of a horizontal tube with a separated flow inside, relative to its lower surface 1-p = 11 MPa; 2-p = 18 MPa; 3-p = 22.4 MPa

by gravitational forces (see Sec. 9.3). Waves which form on flow separation can periodically splash onto the overheated wall. Repeated water splashes cause a sharp cooling of the tube wall. Temperature variations in the metal can cause fatigue phenomena. For this reason, in free-circulation boilers, which are characterized by low velocities of flow in evaporating tubes, horizontal tubes are not heated. In once-through boilers, flow velocities are substantially higher, and therefore, flow separation does not occur within a wider range of heating loads.

The superheating Δt of the upper portion of a tube relative to its lower portion (upon flow separation) can be somewhat diminished by increasing the thickness of the tube wall and the heat conductivity of the tube metal, which causes heat to spread through the tube metal.

At supercritical pressures, the working fluid can separate in horizontal tubes by density in the vertical direction. For this reason, under identical conditions, the temperature of the upper portion of a horizontal tube turns out to be higher than the wall temperature of a vertical tube. The ratio $\alpha_{hor}^{\mu\rho}/\alpha_{ver}$ depends on the enthalpy of the flow and the parameter $q/w\rho$ (Fig. 10.10).

Finned tubes. As has been given earlier, under identical conditions the

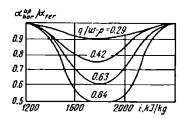


Fig. 10.10. Ratio $\alpha_{hor}^{up}/\alpha_{ver}$ at supercritical pressure (p = constant)

heating load near the front wall on a smooth and a finned tube is roughly the same. This makes it possible to calculate the wall temperature on the front surface of finned tubes of water walls (equal one-sided heating of parallel tubes) as for smooth tubes [see formula (10.12)].

To estimate the reliability of the tube metal, it is essential to know the temperature of metal at the tip of fins. According to [14], with a symmetrical temperature field (the same diameter of welded tubes and the same temperatures of working fluid and heat-transfer coefficients α_2 in the tubes), the temperature at the tips of the fins can be found from the formula:

$$t_{ft} = t_r + k_w \Delta t_f \qquad (10.13)$$

The coefficient k_w considers the effect of welded seams; this can be taken from reference data. The temperature gradient between the root and tip of a fin can be found as:

$$\Delta t_{f} = \mu_{f} \, \frac{q_{\max} H}{\lambda_{fm}} \, \Lambda \quad (10.14)$$

where A is the coefficient of the fin shape depending on the geometrical characteristics h/b_r and b_t/b_r of fins (to be taken from reference data).

The temperature in the root of the fins of a furnace water wall can be expressed by the formula:

$$t_r = t_{ft} + \beta \mu_r q_{\text{max}} \left[\frac{1}{\alpha_2} + \frac{\delta_t}{\lambda_m} - \frac{2}{(1+\beta)} \right]$$
(10.15)

For a platen wall, the temperature of a fin root is found from the formula

$$t_r = t_{f1} + 2\beta \mu_r q_{\max} \left[\frac{1}{\alpha_2} + \frac{\delta_t}{\lambda_m} \frac{2}{(1+\beta)} \right]$$
(10.16)

When it is essential to obtain the mass velocity of the flow, use is often made of multi-pass hydraulic systems in the form of individual banks or sections connected in series so that, as the working fluid flows through them, its temperature and enthalpy gradually increase. On the other hand, these sections are arranged in parallel relative to the heat-transfer agent (combustion products), so that each of them absorbs roughly the same quantity of heat. Under such conditions, an unsymmetrical temperature field can appear in the extreme tubes of adjacent sections. If these tubes are welded together, as is done in gas-tight water walls, the temperature difference between connected fins can give rise to appreciable temperature stresses, sometimes causing fracturing in the welded sections.

Temperature asymmetry in gas-tight (membrane) walls creates the problem of proper temperature distribution in such walls. The problem is solved by applying the superposition principle: with a number of external actions, it is assumed that their effects can be added together. The quantity of heat absorbed by a finned tube consists of two components: the heat absorbed in the cylindrical portion (from the front point to the fin root) and the heat absorbed in the fin proper. Each of these portions has a particular temperature field which can be added together to obtain the total temperature field of a finned tube.

Problems with temperature symmetry (Fig. 10.11a) are solved by assuming that the heat absorbed by the connecting piece between two tubes is distributed evenly between the tubes, and that the maximum of temperature is in the middle of the piece.

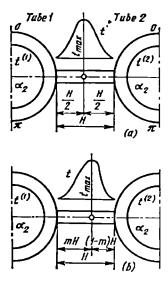


Fig. 10.11. Temperature distribution in a connecting piece

(a) thermal symmetry, $t^{(1)} = t^{(2)}$; (b) asymmetrical conditions, $t^{(1)} > t^{(2)}$

With temperature asymmetry (Fig. 10.11b), the temperature maximum is shifted towards the tube with a lower temperature level, so that the distribution of heat between the tubes will be uneven: it will be divided in proportion to the length of the piece from the point of the highest temperature to a particular tube, i.e. mH for the tube at a higher temperature and (1-m)H for that at a lower temperature, where m is the coefficient of displacement of the temperature maximum; its calculations can be found in reference books [14].

Temperature conditions in a finned tube are determined by its thermal and geometrical parameters. In Fig. 10.12, typical temperature fields are shown in terms of excessive temperatures, i.e. temperature differences between the metal and the working fluid, for a developed half-tube. For all the curves shown, it is typical that the highest temperature is in the front zone of a tube and at the tip of fins. Between these points, the temperature decreases monotonically from the front point to the fin root, then rises sub-

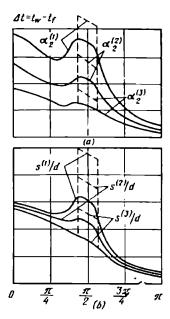


Fig. 10.12. Temperature distribution over the outer surface of a finned tube of water wall (a) at different values of internal heat-transfer coefficient α_2 ($\alpha_2^1 < \alpha_2^2 < \alpha_1^2$); (b) at different relative pitch s/d ($s^1 > s^2 > s^3$)

stantially at the fin tip and again decreases from the fin tip to the rear side of the tube.

The effect of deposits on temperature conditions on heating surfaces. We have discussed temperature conditions on clean (deposit free) heating surfaces. Such conditions can be ensured by proper organization of the processes of steam generation. On the other hand, in boiler operation there is almost always a potential danger that deposits will form from impurities which are present, in a dissolved or suspended state, in water (see Chs. 14 and 15).

When deposits are present on the inner surface of tubes, the wall temperature is found from the formula:

$$t_i^{in} = t_{fl} + q_{in} \left(\frac{1}{\alpha_1} + \frac{\delta_{dep}}{\lambda_{dep}} \right) \quad (10.17)$$

With a larger thickness δ_{dep} of the deposited layer and with a lower heat conductivity λ_{dep} , the thermal re-

sistance across the wall to the working fluid is higher, and thus, the wall will have a higher temperature. The conductivity of deposits depends on the composition of impurities in the water and may vary within very wide limits-from 0.1 to 0.5 W/(m K) for mineral deposits and up to 3-5 W/ /(m K) for ferric oxides. Since the conductivity of deposits is substantially lower than that of the metal, even slight deposits, in a layer of only a few fractions of a millimetre, in the tubes of water walls, which are heated quite intensively, can raise the wall temperature to a value that is inadmissible for reliable operation of the tube metal. Since deposits grow gradually during boiler operation, this circumstance limits the continuous operation of a plant. For this reason, one of the most essential problems in boiler operation is to prevent or restrict the formation of deposits on the steam-water side of heating surfaces (for more detail see Ch. 15).

10.5. Heat Exchange in Steam Generators of Nuclear Power Stations

The unit heating load on the surface of fuel elements is quite high (1 000 kW/m² or even more). This circumstance makes the appearance of poor cooling conditions in fuel elements very probable.

The steam-generating channels of nuclear reactors have an intricate shape. In reactor construction, wide use is made of heating surfaces in the form of bundles of heat-releasing rods between which the heat-transfer agent flows in a longitudinal direction.

The geometry of channels formed between the rods may have a significant effect on the hydrodynamics of the working fluid: the flow rates through the cells of a rod bundle may turn out to be different or the velocity of the working fluid may vary over the cross section of a cell; this may result in the formation of differing cooling conditions over the periphery of the rods. With uneven heat absorption,

which always takes place over the cross section of the core (especially with intensive heating which is typical of nuclear reactors), the differences in the flow rates of the working fluid in the cells and over the periphery of rods may result in different unit heat absorption dq/d ($w\rho$) (per unit flow rate of the coolant) and the appearance of temperature gradients between the rods. Since there is no active turbulent mass exchange of the fluid through the narrow spaces between the rods in the cells, the velocity field of the working fluid may be distorted even more by deformation (bending) of the rods under the action of temperature gradients.

The critical heat flow at which deteriorated conditions of heat transfer can appear on heating surfaces depends on the following principal factors: steam content x, mass velocity wp, and pressure p. For a system of parallel channels in a boiling reactor, these parameters are average if each of the channels receives the same quantity of the working fluid, so that the heat transfer crisis in rod bundles is qualitatively the same as in evaporating tubes, but the quantitative relationships turn out to be substantially different. Considering all the geometry specifics of rod assemblies and the conditions of their operation, differences in unit heat release or heat absorption can result in differences in the flow rate $w\rho$ and steam content x. Since q_{cr} is a decreasing function of x, a boiling crisis is more likely to occur in the space between tubes at a lower limiting steam content than in tubes.

Various types of flow whirlers are mounted in the outlet section of a reactor channel to increase its power; they are often made integral with spacers. Under the effect of whirlers, water droplets are thrown from the flow core onto the walls and replenish the moving liquid film on the walls. Thus, a boiling crisis caused by liquid film drying is delayed and at the outlet the steam-generating reac-

tor can operate at a higher steam content.

In steam generators operating in combination with pressurized water-cooled reactors or sodium-cooled fast-neutron reactors, during evaporation the working fluid passes through the same stages as in organic-fuel steam boilers. The only point of significance is that deteriorated heat transfer in the steam generators of nuclear reactors does not involve failure of the apparatus, since the temperature of its clements cannot exceed that of the heat-transfer agent. However, an increase in the length of the boiling crisis zone, where the intensity of heat transfer is low (see Sec. 10.2), results either in an increase in the steam generator size or in a loss of its capacity.

Moreover, the wall in the zone of liquid film drying is alternately washed with liquid and steam, causing temperature pulsations in the wall (which can be quite substantial in liquid-metal heating) and varying stresses in the metal, which can result in tube cracking.

All these phenomena are probable in steam generators where the working fluid moves in the narrow cells of a channel or in channels of a substantial length.

On the other hand, boiling crisis in the reactor core is extremely dangerous. To prevent its occurrence in water-moderated water-cooled power reactors, the heat-transfer agent of the primary circuit has a temperature a few tens of degrees below the boiling point (subcooling margin).



HYDRODYNAMICS OF OPEN HYDRAULIC SYSTEMS

11.1. Classification of Open Hydraulic Systems

The principal diagrams of open hydraulic circuits in heating surfaces are shown in Fig. 11.1. In any circuit, the reliable operation of the steamgenerating channels depends substantially on stability of motion, i.e. a stable flow rate of the working fluid through parallel tubes and channels. particular conditions (pressure, mass velocity, enthalpy of the working fluid at the inlet, heating intensity) and depending on the design of the steam-generating channels, unstable motion of the working fluid (i.e. with a variable flow rate) can appear in them. One must distinguish between static and dynamic instability. Since instability, per se, is a dynamic process, the term 'static instability' is rather conditional.

Under statically unstable conditions, the flow rates through various tubes (channels) are different and may vary in time with an appreciable frequency. The flow rate in individual tubes may turn out to be insufficient to ensure proper heattransfer conditions. Varying conditions of cooling of the tubes and variations of the wall temperature may cause thermal fatigue of the metal. In some cases, especially in operation at variable or off-design loads, dynamic (or oscillating) instability of motion appears in the form of excessive flow pulsations which lead to variable flow rate of the working fluid,

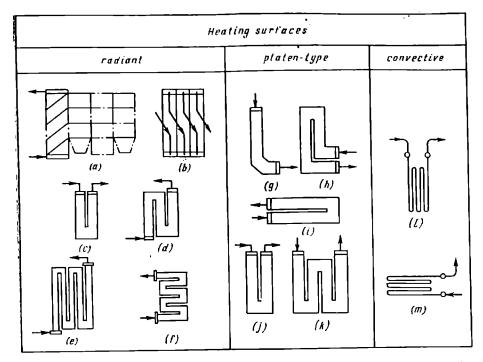


Fig. 11.1. Principal types of hydraulic circuits with forced motion of the working fluid (a) horizontally wound water wall tubes; (b) vertical arrangement of water wall tubes; (c) U-shaped water wall; (d) N-shaped water wall; (e) multi-pass water wall with vertical tubes; (f) multi-pass water wall with horizontal tubes; (g) L-shaped platen; (h) double L-shaped platen; (t) norizontal platen; (f) U-shaped platen; (h) multi-pass vertical platen; (l) vertical convection bank; (m) horizontal convection bank

decreased critical heating load, cyclic temperature variations in the heatod walls and, as a final result, to emergency situations.

Out of all the causes of hydraulic instability, and therefore, of maldistribution of heat, special attention should be given to the effect of an unsteady hydrodynamic characteristic, flow pulsations and headers on, he distribution of the flow between parallel tubes. The pressure gradient in heated tubes can be represented as the sum $\Delta p = \Delta p_{fr} + \Delta p_l + \Delta p_{ac} \pm \Delta p_h$ (see Sec. 9.4). Combining the hydraulic resistances in a single term, $\Delta p_{fr} + \Delta p_l = \Delta p_{hr}$, we obtain:

$$\Delta p = \Delta p_{hr} + \Delta p_{ac} \pm \Delta p_h \tag{11.1}$$

In once-through boilers with horizontal or slightly inclined tubes (Fig.

11.14), the length L of a single tube mny be as high as a few hundred metres and the tube has a large number of bends and therefore, a high hydraulic resistance Δp_{hr} . The hydrostatic component Δp_h of the total pressure gradient in such a circuit is not large in view of the small height of the circuit compared with the length of the developed tube, i.e. $H \ll L$. The resistance due to flow acceleration Δp_{ac} is also not large, especially at a high pressure. Therefore, the total pressure gradient in a circuit with horizontal or slightly inclined tubes is mainly determined by the hydraulic resistance:

$$\Delta p \approx \Delta p_{hr}$$
 (11.2)

Boiler circuits with vertical ascending or ascending-descending motion of the working fluid (Fig. 11.1b to e)

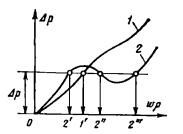


Fig. 11.2. Hydraulic characteristics: (1) stable, (2) unstable

are characterized by a small number of passes, and therefore, by a relatively short tubes and low number of bends. The hydraulic resistance of such circuits is low. The hydrostatic head in vertical tubes, however, makes up a substantial portion of the total pressure gradient, especially at low loads when the contribution of the hydraulic resistance is substantially lower. Thus,

$$\Delta p = \Delta p_{hr} \pm \Delta p_h \tag{11.3}$$

The ratio between Δp_{hr} and Δp_h in the total pressure gradient may have a vital effect on the hydraulic stability of the flow in straight-flow elements. The hydraulic stability of flow is described by the hydraulic characteristic which gives the total pressure gradient Δp as a function of the flow rate G of the working fluid: $\Delta p = f(G)$. If all tubes in an element are of the same diameter, the hydraulic characteristic is a function of the mass velocity: $\Delta p = f(w\rho)$.

The hydraulic characteristic is unambiguous if a particular total pressure gradient in the tubo system has only one corresponding flow rate of the working fluid (curve I in Fig. 11.2), and ambiguous if the same total pressure gradient can appear at two or more different flow rates (curve 2.)

The hydraulic characteristic may become ambiguous on a change in the thermophysical properties of the working fluid, say, the unit volume, which may be caused by a change from one flow rate to another (see Sec. 11.2) or under the effect of hydrostatic head (see Sec. 11.3). The situation is complicated even more by the fact that the effect of hydrostatic head in ascending and descending motion of the working fluid is different. All this results in intricate analytical relationships which are represented graphically for each particular object: hydraulic circuit, geometrical parameters of the tube system, pressure, enthalpy of the working fluid at the inlet, etc.

The hydraulic operation of any circuit is described by its hydraulic characteristic. However, because of the above-mentioned specifics of hydraulic systems with horizontal and vertical tubes, their hydraulic characteristics will be analysed separately.

11.2. Hydrodynamic Stability of Flow in Horizontal Evaporating Tubes

The decisive factor that determines the hydraulic characteristic of evaporating tubes is the temperature of the fluid at the tube inlet. It may be equal or close to the saturation temperature at the pressure at the tube inlet, $t_{ln} \approx t'$, or be substantially lower than that temperature, $t_{ln} < t'$.

If a tube is supplied with subcooled water, steam generation begins not immediately at its inlet, but a certain distance downstream. The whole length of the tube can then be divided into two portions: the economizer portion and the evaporating portion (Fig. 11.3), their lengths being determined by the ratio between the heat release and flow rate of water.

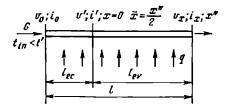


Fig. 11.3. Variations of the working fluid parameters in a heated tube at $t_{in} < t'$

For the same heat release rate, an increase in the water flow rate increases the length of the economizer portion while shortening that of the evaporating portion, which is associated with a lower quantity of steam produced by the heated tube.

If a tube has an economizer portion (Fig. 11.3), we have:

$$\Delta p = \frac{\xi}{2d} (w\rho)^2 \bar{v}_{ec} l_{ec}$$

$$+ \frac{\xi}{2d} (w\rho)^2 \bar{v}_{ev} (l - l_{ec}) \quad (11.4)$$

The unit volume of the working fluid in the economizer and evaporating portions is different. The lengths of these portions are different too.

The unit volume of water varies only slightly along the economizer portion, and therefore, we have:

$$\bar{v}_{ec} = \frac{v_0 + v'}{2} - (11.5)$$

In the evaporating portion, the unit volume of water changes sharply, and it is reasonable to take its mean value in calculations. For tubes that are heated uniformly along their length, quite accurate results are obtained by assuming a linear law of variation of the steam content in the flow, and therefore:

$$\bar{v}_{ev} = v' + \frac{x(v'' - v')}{2}$$
 (11.6)

With uniform heating, the heating load per metre of tube length, $q_l = Q/l$, is constant. The length of the economizer portion will then be found as

$$l_{ec} = (l' - i_0) \frac{G}{Q} l = \frac{\Delta l_{sub} (w\rho) f}{q_l} \quad (11.7)$$

and the length of the evaporating portion:

$$l_{ev} = l - l_{ec} \tag{11.8}$$

Substituting \overline{v}_{ec} , l_{ec} , \overline{v}_{ev} , and l_{ev} into equation (11.4) and arranging the terms according to the powers of $(w\rho)$, we obtain the cubic equation:

$$\Delta p = A (w\rho)^3 - B (w\rho)^2 + C (w\rho)$$
(11.9)

where

$$A = \frac{\xi (v'' - v') \Delta i_{sub}^2 f}{4dq_{t}r}$$
 (11.10)

$$B = \frac{\xi l}{2d} \left[\frac{\Delta l_{sub}}{r} \left(v'' - v' \right) - v' \right] \quad (11.11)$$

and

$$C = \frac{\xi (v'' - v') l^2 q_l}{4 f dr}$$
 (11.12)

If there is no water subcooling at the inlet of an evaporating tube, $\Delta i_{sub} = i' - i_0 = 0$ and the coefficient A turns to zero and B changes sign. Equation (11.9) then takes the f_0 m:

$$\Delta p = B (w\rho)^2 + C (w\rho)$$
 (11.13)

i.e. it is a quadratic equation which expresses unambiguously the hydraulic characteristic.

Equation (11.9) is the equation of the hydraulic characteristic of an evaporating tube at $\Delta i_{sub} > 0$, i.e. in the presence of economizer portion. In this case, the hydraulic characteristic is described by a cubic equation whose solution can have either one real and two imaginary roots, or all three real roots. In the former case, the characteristic $\Delta p = f(w\rho)$ has neither extrema nor common points with a horizontal tangent (curve 1 in Fig. 11.2) and is unambiguous, since any value of pressure gradient Δp has only one corresponding flow rate of the working fluid, $w\rho$. In the latter case, the characteristic has an inflection point and two extrema; it is ambiguous, since any value of pressure gradient Δp has three corresponding different flow rates (curve 2). Ambiguity of the hydraulic characteristic will become clear from the following reasoning. For the same heatrelease rate, as the flow rate of water subcooled under the saturation point increases, the volume velocity of the mixture is not increased, but decreased. An increased flow rate of subcooled water leads to an increase in the length of the economizer portion and in the quantity of heat spent for heating the water to boiling. Accor-

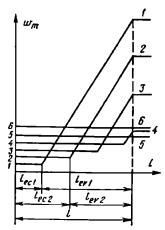


Fig. 11.4. Effect of the flow rate of subcooled water in a steam-generating tube on flow velocity at tube outlet 1-6-water flow rates

dingly, there remains less heat available for evaporation, and therefore, the velocity at the outlet decreases substantially (Fig. 11.4). This continues until the evaporating portion disappears in the tube. As soon as steam generation is interrupted, a further increase in the flow rate of water causes a proportional rise of the velocity along the whole length of the tube, and therefore, a change in the hydraulic resistance of the path. As the water flow rate increases, the hydraulic resistance of the economizer portion, $\Delta p_{\sigma c}$, increases while that of the evaporating portion, Δp_{ev} , decreases. Depending on the ratio between Δp_{ec} and Δp_{ev} , the total hydraulic resistance of the path may either increase or decrease with an increase of the load in a particular range of flow rates, resulting in the hydraulic characteristic being either unambiguous (steady-state) or ambiguous (unsteady-state).

Let us consider the hydraulic characteristic of a once-through element, which has three real roots by equation (11.9), see Fig. 11.5. At vory low flow rates of water, $(w\rho) < (w\rho)$, and superheated steam will form at a given heat release rate, since the path

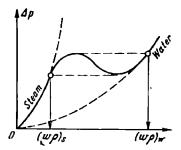


Fig. 11.5. Region of the hydraulic characteristics of a tube producing steam-water mixture

has neither an economizer nor evaporating portion, but is essentially a superheater along its whole length. At very high flow rates of water, $(\omega\rho) > (\omega\rho)_{\omega}$, the available heat is insufficient to heat the water to the saturation point, there is no steam generation, and the whole path is essentially an economizer which delivers water. For these extreme cases, the hydraulic characteristic is described by the parabolic equation:

$$\Delta p = \xi \frac{(w\rho)^2}{2} v$$
 (11 14)

As follows from this equation, with the same mass velocity of the flow, the resistance to the motion of steam is greater than that to the motion of water, since v_s is greater than v_w .

If a steam-water mixture forms in the path [the range of flow rates $(w\rho)_{\bullet} < (w\rho) < (w\rho)_{w}$], the characteristic may become unstable and the flow rates may vary, so that a steamwater mixture with widely different steam contents will be delivered periodically. Some of these steam contents may turn out to be excessive for the given heat release rate, meaning that the cooling of the tubes will become unreliable.

An unstable hydraulic characteristic can result in different flow rates through individual parallel tubes that are connected to a common header. A more dangerous situation, however, results when the flow rate through a tube changes. For the same heat re-

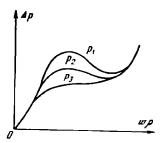


Fig. 11.6. Effect of pressure on the stability of hydraulic characteristic $(P_1 < P_2 < P_3)$

lease rate, this can cause temperature variations in the tube wall and the appearance of critical fatigue stresses and, eventually, damage of the evaporating tubes.

Since the ambiguous hydraulic characteristic is mainly due to a large difference between the unit volumes of water and steam, an increase in pressure can make the characteristic more stable and the motion of the working fluid more steady-state (Fig. 11.6).

As noted earlier, instability of the hydraulic characteristic is associated with the presence of an economizer portion in a tube. For this reason, with a higher degree of water subcooling at the tube inlot, the characteristic is less stable (Fig. 11.7). On the other hand, if subcooling is decreased by raising the inlet temperature close to t, another problem may result: as the steam-water mixture enters the inlet header of a bank of evaporating tubes, it will be separated into steam

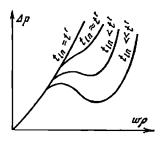


Fig. 11.7. Effect of water subcooling at the inlet of a steam-generating tube on the stability of hydraulic characteristic (p constant)

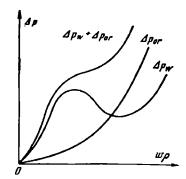


Fig. 11.8. Transformation of an unstable hydraulic characteristic into stable by throttling subcooled water flow

and water in the header, so that some tubes will receive more water and less steam than others, i.e. the flow rates of water and steam in them will be sharply different and uncontrollable; this can lead to overheating, and even burn-through of the evaporating tubes which are heated quite intensively in the boiler furnaces.

For the above reasons, once-through boilers are made with a non-boiling economizer section. No-boiling conditions must be ensured not only at the rated load, but also in operation under any load and with any kind of fuel for which the boiler has been designed.

An unstable hydraulic characteristic can be stabilized by introducing an additional resistance in the economizer portion of the hydraulic system, which will vary by a parabolic law depending on the flow rate of the working fluid. As shown in Fig. 11.8, the initial and the additional characteristic are added graphically so that the resulting characteristic is stable. The hydraulic resistance of the economizer portion is usually increased by two methods: by increasing its resistance locally (mounting an orifice plate) or at an appreciable length (providing a stepped tube).

Orifice plates. Orifice plates produce a pressure gradient proportional to the square of the flow rate of the single-phase fluid that passes through them:

$$\Delta p_{or} = \xi_{or} - \frac{(w\rho)^2}{2} v' = \Psi_{or} (w\rho)^2$$
 (11.15)

where $\xi_{\sigma r}$ is the coefficient of resistance of the orifice plate; $\Psi_{\sigma r} = \xi_{\sigma r} (v'/2)$.

Orifice plates are mounted at the inlet to evaporating tubes which have an economizer portion, i.e. in a place where the flow is single-phase.

Noting equations (11.9) and (11.15), the total resistance of a tube with an orifice plate is described by the equation:

$$\Delta p = A \ (w\rho)^3 + (\Psi_{\sigma r} - B) \ (w\rho)^2 + C \ (w\rho)$$
(11.16)

Unambiguity of the total characteristic is not a sufficient condition, since slight variations in the pressure gradient can cause substantial changes in the flow rate (Fig. 11.9). For this reason, the total characteristic should have a certain degree of steepness, i.e. the relative variation in the flow rate of the working fluid should exceed the relative variation of the pressure gradient by at least three times.

Stepped tubes. The required degree of throttling of the flow to obtain a stable hydraulic characteristic can also be achieved by increasing the hydraulic resistance of the coils in the economizer proper. For this purpose, the tubes in the economizer portion

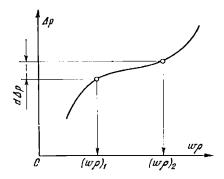


Fig. 11.9. Effect of insufficiently stable characteristic on the water flow rate in a steamgenerating tube

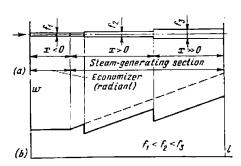


Fig. 11.10. A stepped tube (a) and variations of flow velocities in it (b)

are of a smaller diameter than in the evaporating portion, i.e. a stepped tube is employed. This increases the flow velocity and hydraulic resistance in the economizer portion where the unit volume of the working fluid is relatively low, and diminishes the velocity and hydraulic resistance in the evaporating portion where the steam content of the mixture increases gradually (Fig. 11.10).

The hydraulic characteristic of horizontal evaporating coils at supercritical pressures can also be ambiguous. Its ambiguity is determined by the unit enthalpy and the unit volume of water at the inlet. When the water enthalpy at the inlet is substantially lower than the enthalpy of phase transition, the characteristic will be ambiguous. When i_{in} increases, it becomes more stable. For this reason, hydraulic characteristics are calculated for the minimum inlet enthalpy of water that is probable in operation. In the region of supercritical pressures, an increase in inlet pressure can also improve the stability of the hydraulic characteristic.

11.3. Hydrodynamic Stability of Flow in Vertical Evaporating Tubes

In vertical tube walls with ascending or ascending-descending motion of the working fluid and a small number of passes (II-, U- and N-shaped sections, Fig. 11.1) whose height is comparable with the length of the

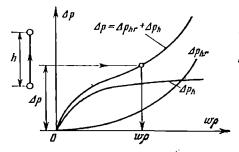


Fig. 11.11 Hydraulic characteristic of a vertical uptake tube

developed tube, the hydraulic characteristic is also determined mainly by the pressure and the degree of water subcooling at the inlet. The hydraulics in such tubes are strongly influenced by the hydrostatic head Δp_h . The hydrostatic component of the total pressure gradient is substantial both at subcritical and supercritical pressures. There is no principal difference in the magnitude of Δp_h in these cases, but Δp_h has a stronger effect at subcritical pressures.

In a single-pass vertical uptake section (Fig. 11.11), the working fluid has to overcome hydraulic resistance and hydrostatic head, so that both these terms are positive in formula (11.3) and the pressure gradient between the upper and lower header of the section is determined as their sum:

$$\Delta p_h^{\prime} = \Delta p_{hr} + \Delta p_h \qquad (11.17)$$

For a given heat release rate and a flow rate close to zero, the tubes are filled with steam which has a low density, and therefore, the hydrostatic component in this zone can be neglected.

As follows from Fig. 11.11, the hydraulic characteristic of such a section is stable, that is, each pressure gradient has a single corresponding definite flow rate.

With descending motion of the working fluid in tubes (Fig. 11.12), the hydraulic resistance Δp_{hr} is positive while the hydrostatic head in formula (11.3) is negative, since it

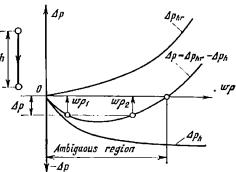


Fig. 11.12. Hydraulic characteristic of a vertical downtake tube

acts in the direction of fluid motion. Therefore, the pressure gradient between the headers is determined as the difference of these terms:

$$\Delta p! = \Delta p_{hr} - \Delta p_h \qquad (11.18)$$

With descending motion, the flow rate of the working fluid may be unstable, which means that a single value of the pressure gradient may be associated with two widely different flow rates. The zone of ambiguity in real tube sections may involve a wide range of flow rates [up to 1 000-2 000 kg/(m² s)].

By comparing Figs. 11.11. and 11.12, we see that Δp_h improves the hydraulic characteristic of an ascending flow and adversely affects that of a descending flow.

In tube systems with ascendingdescending motion (II-shaped sections), the hydraulic resistance is overcome in both directions, and therefore, is determined as the sum of resistances in the ascending and descending portions: $\sum \Delta p_{hr} = 1 \Delta p_{as} +$ $+ \Delta p_{des}$. The hydrostatic component for the ascending element, Δp_h^{as} (curve Δp_h^{as} in Fig. 11.13a) is similar to that shown in Fig. 11.11, since the enthalpy of fluid at the inlet, iin, has been assumed, for simplicity, to be independent of the flow rate, i.e. constant. The beginning of the descending element of a tube is the continuation of the preceding ascending

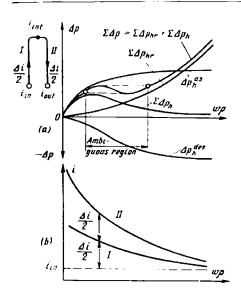


Fig. 11.13. Hydraulic characteristic of a Il-shaped heated tube

element in which enthalpy i_{int} varies according to the flow rate at a given heat release. With a smaller flow rate of the working fluid, i_{int} is higher (curve I in Fig. 11.13b), so that the enthalpy of the working fluid sharply increases in the descending element of the tube (curve II), especially in the outlet portion. With a decrease in the flow rate, this results in an essential decrease in the flow density and, accordingly, in the hydrostatic component Δp_{i}^{her} .

Adding together the hydrostatic components for the ascending and descending element, Δp_h^{ns} and Δ_h^{der} , we

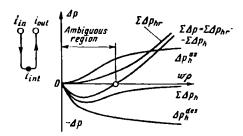


Fig. 11.14. Hydraulic characteristic of a U-shaped heated tube

obtain the hydrostatic head $\sum \Delta p_h$ in a Π -shaped section.

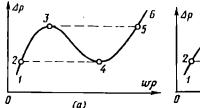
The total hydraulic characteristic of a II-shaped section

$$\Sigma \Delta p = \Sigma \Delta p_{hr} + \Sigma \Delta p_{h} \quad (11.19)$$

may be ambiguous in a wide range of flow rates of the working fluid (Fig. 11.13a).

Figure 11.14 shows the total hydraulic characteristic and its components for a U-shaped section, which has been constructed according to the method discussed above. The hydraulic resistances in the ascending and descending branches are positive and are thus added together $(\Sigma \Delta p_{hr})$. The hydrostatic component in the descending branch, Δp_h^{drs} , is negative and the onthalpy of fluid at the inlet to this branch, i_{in} , is assumed to be constant and independent of the flow rate. The beginning of the ascending motion coincides with the end of the descending motion. The hydrostatic component of the ascending motion is positive, but the enthalpy at the inlet to the ascending portion (in the point of transition from descending to ascending flow), i_{int} , is dependent on the flow rate. At low flow rates, i_{In} is high and the density of the fluid is low, which has a retarding effect on the rise of the hydrostatic head with the increasing flow rate. in that zone. Only in the zone of large flow rates at which the quantity of heat per unit flow rate of the working fluid is not as sub**st**antial, i_{int} becomes lower and the hydrostatic head Δp_h^{as} becomes much larger. The resulting curve of hydrostatic heads is denoted as $\sum \Delta p_h$ in the figure. As may be seen, the total hydraulic characteristic of a U-shaped section is ambiguous in a wide range of flow rates of the working fluid.

Hydraulic characteristics of N-shaped and more complicated multi-pass systems can be obtained in a similar manner. As follows from the above analysis, the hydrostatic pressure gradient improves the hydraulic characteristic of an ascending flow and ad-



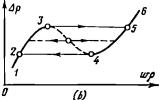


Fig. 11.15. Ambiguous hydraulic characteristics (a) of a single tube: (b) of a tube system

versely affects that of a descending flow. In this respect, a U-shaped circuit is better than a Il-shaped, since the working fluid in its outlet portion, where it has a high steam content, and therefore, lower density, moves upwards and the effect of the hydrostatic head is positive. The hydraulic characteristic is improved even more in an N-shaped system with an inlet header arranged at the top and an outlet header at the bottom. In addition, this system has two ascending portions per one descending. In general, tube sections with a low number of passes have either an ambiguous or insufficiently stable hydraulic characteristic.

With an increasing number of passes in a hydraulic system, the effect of hydrostatic component on the total pressure gradient diminishes, but the role of hydraulic resistance becomes more significant. When the number of passes is greater than 8-10, the hydraulic characteristic of a multipass system approaches that of the horizontal evaporating tubes.

An unstable hydraulic characteristic of a single tube (channel) with ascending or descending motion of the working fluid under the action of hydrostatic head is realized in the whole region of ambiguity (curve 2-3-4-5 in Fig. 11.15a), since the flow rate is determined by external conditions, i.e. by pump capacity. Real heating surfaces consist of a large number of parallel tubes. In multitube systems, the total flow rate of the fluid is also determined by pump

capacity, but the flow rates through individual tubes may turn out to be different. As has been established by experiment and confirmed by experience, hydraulic characteristics in such systems are realized only in ascending portions (the leftmost curve 1-2-3 and the rightmost curve 4-5-6 in Fig. 11.15b). A decrease in the flow rate in a tube near a minimum of the hydraulic characteristic (at point 4) will cause a sudden drop in the flow rate and a change in the conditions described by point 2. A rise in the flow rate at a maximum of the characteristic (near point 3) can lead to a sudden increase in the flow rate in some tubes of the circuit and a change in the conditions of point 5. The descending branch of the characteristic (portion 3-4) is not realized, except for cases where the system does not have many tubes (three or four). In this region of flow rates, unambiguous motion cannot be ensured with the result that the flow rate either jumps into the region of lower values (left-hand branch) or into that of higher values (right-hand branch), while the total flow rate of the working fluid remains unchanged.

Operation on the left-hand branch of the characteristic usually cannot maintain the required temperature conditions of intensively heated evaporating tubes. The sole real region of operation of a once-through circuit is the right-hand branch of the characteristic which includes ambiguous and unambiguous portions. In the unambiguous portion 5-6 of that branch,

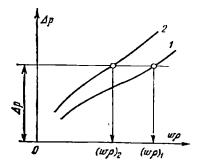


Fig. 11.16. Effect of maldistribution on the working fluid flow rate in parallel tubes with stable hydraulic characteristics

I - hydraulic characteristic of an element; 2-hydraulic characteristic of a malfunctioning tube

the velocity of the working fluid is so high that it often turns out to be unfeasible. Thus, one must operate on the ambiguous portion 4-5 for which the boundaries of reliability should be determined. In circuits composed of parallel tubes of the same design and heated identically, stability in the region of an ambiguous hydraulic characteristic can be ensured for the entire external (ascending) branch of the characteristic (portion 4-5 in Fig. 11.15b).

In a system of parallel tubes, heat absorption is always different in view of the structural and hydraulic nonidentity of the tubes (see Sec. 11.4). Therefore, some tubes whose operating conditions deviate from the average values for the system (malfunctioning tubes) have a different hydraulic characteristic than is average for the system. As seen from Fig. 11.16, even with an unambiguous characteristic of the system, different (but stable) flow rates of the working fluid may appear in a reference tube and a malfunctioning tube.

With parallel operation of a system of tubes with different ambiguous characteristics, three cases are possible: the pressure gradient between the header of the system at the minimum of the hydraulic characteristic for a malfunctioning tube can be either (a) smaller than the pressure gradient in the circuit (Fig. 11.17a),

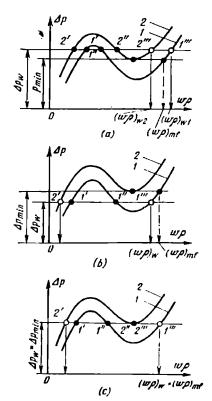


Fig. 11.17. Hydraulic characteristics of IIshaped circuits 1—element; 2—malfunctioning tube

(b) greater than that pressure gradient. (Fig. 11.17b), or (c) equal to the pressure gradient in the circuit. These cases are analysed with a gradually decreasing total flow rate of the working fluid, and therefore, a decreasing pressure gradient between the headers of the hydraulic system.

In the first case, when the pressure gradient in the system exceeds the pressure gradient at the minimum of the characteristic of the worst malfunctioning tube, $\Delta p_w > \Delta p_{\min}$, i.e. of the malfunctioning tube whose minimum of the characteristic is higher than the minimum of the characteristic of the main number of tubes, three operating points are possible on each of the curves (1', 1" and 1" on the general characteristic of the main number of tubes and 2', 2" and 2"

on the characteristic of the malfunctioning tube). Points 1" and 2" on the descending branches of the characteristics cannot be realized (see Fig. 11.15b). Points I' and I' on the lefthand ascending branch of the characteristic are also not realizable as long as the condition $\Delta p_{10} > \Delta p_{min}$ is satisfied. This means that unfavourable conditions in an element with a low flow rate of the working fluid can be prevented by providing a flow rate at which the pressure gradient in the characteristic of the main number of tubes will be higher than the pressure gradient at the minimum of the hydraulic characteristic of a malfunctioning tube, $(w\rho)_{\omega 1} > (w\rho)_{mf}$.

With $\Delta p_{\omega} < \Delta p_{\min}$, there are no working points on the descending and right-hand ascending branches of the characteristic of a malfunctioning tube. Under such conditions, the only remaining points which correspond to the two different flow rates of the working fluid are 2' and 1". At one of them (point 1"), a high flow rate is established, which ensures stable operation of the majority of the tubes in the circuit. At point 2', the flow rate is too low for proper cooling of the worst malfunctioning tube, with the result that the tube system will be unreliable.

With $\Delta p_w = \Delta p_{\min}$, the flow rate at the minimum of the characteristic of a malfunctioning tube is unstable. In this case, a pressure gradient in the tube system has two different corresponding flow rates. One of them, in the malfunctioning tube, has a low value, 2', which cannot ensure proper cooling conditions. The other flow rate, 1" on the characteristic of the majority of the tubes, ensures proper cooling of these tubes. The flow rates in points I' and I'' for an average element are unreal for the reasons given earlier. The condition $\Delta p_w = \Delta p_{\min}$ determines the boundary of stability of a malfunctioning tube, i.e. the extreme state up to which the operation of the whole circuit can be stable.

As regards the stability conditions, the lowest mass velocity allowed in a system is expressed by the inequality

$$(\omega \rho)_{\omega} \geqslant m \ (\omega \rho)_{mt} \tag{11.20}$$

where $(wp)_{mf}$ is the mass velocity in a system of tubes (channels) on an external branch of its hydraulic characteristic in the point corresponding to the minimum of the hydraulic characteristic of a malfunctioning tube, $(wp)_w$ is the mass velocity in the system, and m is the reserve factor, which is assumed to be equal to 1.5.

Stable operation in the region of ambiguous hydraulic characteristics can be ensured by throttling the flow by means of orifice plates or stepped tubes. Another method is to carefully select the mass volocity wo of the working fluid at the rated heating load of a particular water-wall section so as to ensure reliable cooling of the tubes at a given minimal heating load in view of their thermal and structural non-identity, that is, to run the circuit on an external ascending branch of its hydraulic characteristic, $d(\Delta p)/d(w_0) > 0$. The latter method is preferable, since it does not require tube throttling. In most cases, orifice plates are employed when it is necessary to equalize the flow rates of the fluid through parallel hydraulic elements, such as sections of furnace water walls. In some cases, flow stability can be achieved more efficiently by changing over to another scheme of tube connection which has no downtake tubes.

The stability of the hydraulic characteristic can be affected not only by heated (evaporating) tube sections, but also by unheated steam-circulating tubes. For instance, Fig. 11.18 shows an unlucky arrangement with external unheated steam-circulating tubes connected to a lower header. The hydrostatic head in the tubes is negative and the circuit is essentially a II-shaped system with an ambiguous hydraulic characteristic. With hydraulic maldistribution and too low working fluid velocities (due to a low

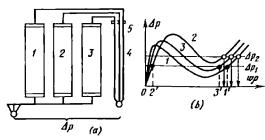


Fig. 11.18. II-shaped circuit with unheated downtake tubes (a) and its hydraulic characteristics (b)

1-3--parallel heated water walls and their respective hydraulic characteristics; 4-unheated downtake tubes; 5--upper mixing header

gradient Δp_1), ambiguity of the characteristic may result in section 2 malfunctioning and having a low flow rate on the left-hand branch 2' of the characteristic. Thus, under completely favourable operating conditions in sections 1 and 3 (the corresponding flow rates are denoted as 1' and 3'), section 2 will operate unreliably. Reliable operation in the circuit can be achieved by providing a pressure gradient Δp_2 at which the working points will be on the extreme branches of the characteristics of all the sections and will have close velocity values. It is also possible to eliminate ambiguity of the hydraulic characteristics of the sections by providing a mixing header at their top.

The operation of complex circuits with a number of once-through elements is analysed by using their resulting hydraulic characteristics. These are obtained graphically by adding the characteristics of individual elements together, taking account of the scheme of their connection. If some elements are connected in series, the flow rate of the working fluid is the same in all of them. For this reason, the resulting characteristic is const-

ructed by adding together the characteristics of the elements at the same flow rate of the working fluid (Fig. 11.19a). The hydraulic resistance of such a system is higher than the resistance of anyone of its elements. In complex once-through circuits composed of a number of elements connected in parallel, the hydraulic resistances of the elements are constant. The general characteristic of the system is obtained by adding together the characteristics of the elements taken at the same pressure gradient in the circuit (Fig. 11.19b). In such a case, the hydraulic resistance of the circuit is smaller than that of anyone of its elements.

11.4. Maldistribution of Heat

For reliable operation of a heating surface (section), it is extremely important that all its parallel tubes operate under rated (average) conditions. In practice, however, one has to bear in mind that the tubes of a section may have different hydraulic characteristics (due to variations in their diameter, length, surface roughness, effect of headers, or effect

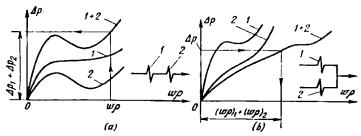


Fig. 11.19. Summation of hydraulic characteristics for series connection of ouce-through elements (a) and for their parallel connection (b)

of instability in operation of evaporating tubes) and different thermal characteristics (different heat absorption due to their different arrangement relative to the flow of combustion products, uneven slagging, fouling, etc.).

Differences in the hydraulic and thermal characteristics of tubes are especially evident in high-capacity plants with developed heating surfaces where deviation of some elements from the rated conditions are more probable. Under such conditions, the working fluid may be distributed unevenly between the tubes, and therefore, its unit enthalpy at the outlet of some tubes may differ substantially from the average value. In some malfunctioning tubes, dangerous temperature conditions may appear.

The reliable operation of a oncethrough element can be characterized by two groups of factors:

(a) parameters of an element that operates under average conditions in the system:

$$t_{el}^{ex}$$
, Δi_{el} , G_{el} , q_{el} , z_{el} , H_{el}

(b) parameters of a malfunctioning tube (or channel):

$$t_{mf}^{ex}$$
, Δi_{mf} , G_{mf} , q_{mf} , z_{mf} , H_{mf}

where G is the flow rate of the working fluid in an element, q is the quantity of heat transferred to it, Δi is the heat absorbed by an element, H is the area of a heating surface, z is the coefficient of hydraulic resistance, and t^{nx} is the temperature of the working fluid at the outlet from parallel tubes; the subscripts 'el' and 'mf' stand respectively for an element under average conditions and a malfunctioning tube.

Let us introduce some additional designations:

$$\rho_h = G_{ml}/G_{el} \tag{11.21}$$

is the coefficient of hydraulic maldistribution;

$$\rho_a = \Delta i_{ml} / \Delta i_{el} \qquad (11.22)$$

is the coefficient of maldistribution of heat:

$$\eta_{ha} = q_{mt}/q_{ct} {11.23}$$

is the coefficient of non-uniform heat absorption;

$$\eta_h = z_{mf}/z_{el}$$
 (11.24)

is the coefficient of hydraulic nonuniformity;

$$\eta_{str} = H_{mt}/H_{el}$$
 (11.25)

is the coefficient of structural non-identity.

Taking the coefficient of maldistribution of heat as a basis, we can establish the relationship between the coefficients indicated:

$$\rho_{q} = \frac{\Delta i_{mf}}{\Delta i_{el}} - \frac{q_{mf}H_{mf}}{G_{mf}} : \frac{q_{el}H_{el}}{G_{el}}$$
(11.26)

or, on substituting the corresponding values from (11.21), (11.23) and (11.25):

$$\rho_q = \frac{\eta_{ha}\eta_{str}}{\rho_h} \tag{11.27}$$

The coefficient of structural nonidentity η_{str} is not associated with the processes which occur in a tube system; it is usually estimated as $\eta_{str} = 0.95$ -1.

The coefficients of thermal and hydraulic maldistribution are interrelated as follows:

$$\rho_a = \eta_{ha}/\rho_h \qquad (11.28)$$

Maldistribution of heat may be caused by different thermal characteristics of parallel tubes, while hydraulic maldistribution is due to differences in their hydraulic characteristics. As follows from equation (11.28), maldistribution of heat can be caused either by different heat absorption, by hydraulic maldistribution, or by both. Maldistribution of heat depends not only on the extent of thermal nonuniformity and hydraulic maldistribution, but also on a particular combination of these two factors. Tubes which are heated the most intensively and, on the other hand, with the lowest flow rates through them, turn out to be operating under the most dangerous conditions. If the greatest differences of various kinds (in heat absorption, flow rate, structural differences, etc.) are found not in a singletube, but in various tubes, they all should be checked separately for maldistribution of heat. Those with the worst maldistribution of heat should be checked additionally for long-term strength and scaling conditions.

Hydraulic maldistribution. In a system of parallel tubes connected to headers at the inlet and outlet and with forced motion of the working fluid, measures should be taken to distribute the fluid evenly between the tubes. Under practical conditions, the distribution of flow rates is always more or less uneven, i.e. hydraulic maldistribution occurs [see formula (11.21)]. Hydraulic maldistribution may appear due to hydraulic non-identity of the parallel tubes, which may be caused by differences in their hydraulic resistance, due to the header effect, or through pressure variation along the length of the header (see Sec. 11.5). This kind of maldistribution appears mainly in the heating surfaces of superheaters and, to a lesser extent, in economizers.

To derive the equation for hydraulic maldistribution, let us write the equation for total pressure gradient in an element:

$$\Delta p_{el} = z_{el} - \frac{(w\rho)_{el}^2}{2} \bar{v}_{el} + (w\rho)_{el}^2 (v_l^{el} - v_{in}^{el}) + \overline{\Delta p_{head}^{el}} + \Delta p_h^{el}$$
 (11.29)

and similarly for a malfunctioning tube

$$\Delta p_{mf} = z_{mf} - \frac{(w\rho)_{mf}^2}{2} \bar{v}_{mf} + (w\rho)_{mf}^2 (v_f^{mf} - v_{in}^{mf}) + \overline{\Delta} p_{head}^{mf} + \Delta p_h^{mf}$$
(11.30)

When there is a large number of parallel tubes, the effect of a change in heat absorption in a malfunctioning tube on the total pressure gradient can be neglected, and therefore, $\Delta p_{el} = \text{constant}$. Since the pressure gradient between the inlet section of the supply header and the outlet section of the discharge header is the same for all the tubes which are con-

nected in parallel in the system, it can be written that $\Delta p_{mf} = \Delta p_{el}$.

Solving simultaneously equations (11.29) and (11.30), we obtain the expression for hydraulic maldistribution:

$$\rho_{h} = \frac{(w\rho)_{mf}}{(w\rho)_{el}}$$

$$\sqrt{\frac{z_{el}v_{el} + 2(v_{f}^{el} - v_{in}^{el})}{z_{mf}v_{mf} + 2(v_{f}^{mf} - v_{in}^{mf})}}}$$

$$\times \left(1 - \frac{\delta\Delta p_{head} + \delta\Delta p_{h}}{\Delta p_{hr} + \Delta p_{ac}}\right) \quad (11.31)$$

where $\delta\Delta p_{head} = \Delta p_{head}^{mf} - \Delta p_{head}^{cl}$ is the difference of pressure gradients in the headers between the sections with a malfunctioning tube and an element, $\delta\Delta p_h = \Delta p_h^{ml} - \Delta p_h^{el}$ is the difference of hydrostatic heads in the malfunctioning tube and element, Δp_{hr} and Δp_{ac} are taken for the element.

Hydraulic maldistribution in superheaters and non-boiling economizers. The formula for the coefficient of hydraulic maldistribution acquires the simplest form for non-boiling economizers in which the working fluid in a horizontal or ascending motion, has an ample reserve of subcooling. In this case: $v_{el} \approx v_{el} = \text{constant};$ $v_{el} \approx v_{mf}$; the difference of hydrostatic heads between a malfunctioning tube and element and the resistance due to flow acceleration are negligible: $\delta \Delta p_h \approx 0$, $\Delta p_{ac} \approx 0$. With a uniform water supply to and removal from the headers, the coefficient of hydraulic maldistribution for these conditions is determined by the ratio of the coefficients of hydraulic resistance:

$$\rho_h = \sqrt{\frac{z_{el}}{z_{mf}}} = \frac{1}{\sqrt{\eta_h}} \quad (11.32)$$

With the working fluid supplied to and removed from a header through its end faces, the header effect is substantial and the formula for hydraulic maldistribution takes the form:

$$\rho_h = \sqrt{\frac{1}{\eta_h} \left(1 - \frac{\delta \Delta p_{head}}{\Delta p_{hr}}\right)} \quad (11.33)$$

In non-boiling convective economizers with horizontal tube coils, the coefficient of hydraulic maldistribution usually does not exceed 0.9. If downtake motion takes place in a non-boiling economizer and the hydraulic characteristic may be ambiguous, the hydrostatic component should be taken into account; this is also necessary for all boiling economizers.

Hydraulic maldistribution in superheaters heavily depends on differences in the total coefficients of tube resistance, and in their heat absorption, and on pressure changes along the headers. Neglecting the pressure variation caused by steam flow accelera-

tion, we get:

$$\Phi_{h} = \sqrt{\frac{1}{\eta_{h}} \frac{\bar{v}_{el}}{\bar{v}_{mf}} \left(1 - \frac{\delta \Delta p_{head} + \delta \Delta p_{h}}{\Delta p_{hr}} \right)}$$
(11.34)

The header effect should be considered when $\delta \Delta p_{head}/\Delta p_{hr} \gg 0.05$. The hydrostatic component of the total pressure gradient is substantial only in single-pass superheaters at $\delta \Delta p_h / \Delta p_{h\tau} \gg 0.05$.

Hydraulic maldistribution in the zones of sudden changes in the unit volume of the working fluid. In modern once-through boilers, steam-generating elements are usually arranged in the lower radiation section which is heated

the most intensively.

The combination of a substantial hydraulic maldistribution and intensive heating can, under certain conditions, result in a sharp rise in temperature and unit volume of the working fluid in a malfunctioning tube, reduction in the flow rate through it, and overheating of the tube metal until burn-out occurs [35]. Hydraulic maldistribution in such tubes may appear on an appreciable change in the unit volume of the heated working fluid. It appears mainly in the heating surfaces of the high-heat capacity zones of supercritical-pressure boilers, in the evaporating surfaces of subcritical-pressure boilers, and in boiling economizers.

In some cases, the hydraulic characteristic is stabilized by mounting orifice plates at the entry to tubes. In steam generation zones, the pressure losses Δp_{ac} and Δp_{head} are usually neglected. For such conditions, and noting the assumptions made earlier, formula (11.31) takes the form:

$$p_h = \sqrt{\frac{z_{el} \bar{\nu}_{el} + \xi_{or}^{el} \nu_{or}^{el}}{z_{ml} \bar{\nu}_{mf} + \xi_{nr}^{ml} \nu_{or}^{ml}} \left(1 - \frac{\delta \Delta p_h}{\Delta p_{hr}}\right)}$$
(11.35)

where ξ_{or}^{cl} , ξ_{or}^{ml} , v_{or}^{cl} , v_{or}^{ml} are the coefficients of resistance of the orifice plates and unit volumes of the fluid flowing through the orifice plates mounted in an element and malfunctioning tube.

The difference in hydrostatic heads between a malfunctioning tube and an element can be written in the following form:

$$\begin{split} \delta\Delta\rho_{h} &= \Delta\rho_{h}^{mf} + \Delta\rho_{h}^{el} \\ &= \{ [\Sigma \left(\bar{h\bar{\rho}} \right)_{as}^{mf} + \Sigma \left(\bar{h\bar{\rho}} \right)_{des}^{mr}] \\ &= \{ [\Sigma \left(\bar{h\bar{\rho}} \right)_{as}^{el} + \Sigma \left(\bar{h\bar{\rho}} \right)_{des}^{el}] \} g \\ &= \{ [\Sigma \left(\bar{h\bar{\rho}} \right)_{as}^{mf} + \Sigma \left(\bar{h\bar{\rho}} \right)_{des}^{el}] \\ &= [\Sigma \left(\bar{h\bar{\rho}} \right)_{des}^{mr} + \Sigma \left(\bar{h\bar{\rho}} \right)_{des}^{el}] g \end{split}$$
(11.36)

Water walls of steam boilers in high-capacity monobloc units are made in the form of vertical sections, horizontally coiled sections, U- and Nshaped sections, multi-pass sections with horizontal tubes, etc. Hydraulic maldistribution in some of these types will be discussed below.

Horizontal circuit. In horizontal tubes there is no hydrostatic head, and therefore, the formula of hydraulic maldistribution with throttling takes the form:

$$\rho_h = \sqrt{\frac{z_{el} \bar{\nu}_{el} + \xi_{or}^{vl} v_{or}^{vl}}{z_{mf} \bar{\nu}_{mf} + \xi_{or}^{ml} v_{or}^{ml}}}$$
(11.37)

If there is no throttling, the formula is simplified:

$$\rho_h = \sqrt{\frac{z_{el}\bar{v}_{el}}{z_{ml}\bar{v}_{ml}}} = \sqrt{\frac{1}{\eta_h} \frac{\bar{v}_{el}}{\bar{v}_{ml}}}$$
(11.38)

Vertical circuit. In vertical circuits, the hydrostatic head has a vital effect on hydraulic maldistribution. In some particular cases, and taking into consideration the effect of various factors (in formula (11.36)), the equation for hydraulic maldistribution (11.35) takes the form:

single-pass ascending section:

$$\sqrt{\frac{1}{\eta_{h}} \frac{\bar{v}_{el}}{\bar{v}_{mf}}} = \sqrt{\frac{1}{1 + \frac{2gh(\bar{\rho}_{el} - \bar{\rho}_{mf})}{z_{en}\bar{v}_{el}(w0)_{el}^{2}}}}$$
(11.39)

single-pass descending section:

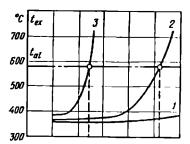
$$V = \frac{\sqrt{\frac{1}{\eta_h} \frac{\bar{v}_{el}}{\bar{v}_{ml}}}}{1 - \frac{2gh(\bar{\rho}_{el} - \bar{\rho}_{ml})}{z_{el}\bar{v}_{el}(w\rho)_{el}^2}}$$
(11.40)

two-pass U-shaped section:

As follows from formula (11.38), the coefficient of hydraulic maldistribution in horizontal tubes depends on the ratio between the thermal and the hydraulic characteristics in an element and a malfunctioning tube.

In vertical tubes, hydraulic maldistribution also depends heavily on the hydrostatic component and its contribution to the total pressure gradient between the discharge and supply headers, i.e. the ratio between Δp_h and Δp_{hr} .

On the other hand, the relation between the increments of enthalpies in a malfunctioning tube and an average element is characterized by the coefficient of maldistribution of heat ρ_q [see formula (11.22)]. Assuming a number of values of $\rho_q = \Delta i_{mf}/\Delta i_{et}$, one can determine the coefficient of hydraulic maldistribution $\rho_h = (w\rho)_{mf}/(w\rho)_{et}$



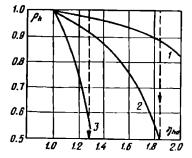


Fig. 11.20. Maldistribution characteristics of a once-through element at $p\approx 24$ MPa and $i_{In}=1.200$ kJ/kg Δi_{el} kJ/kg: I=400; z=600; z=1.000

and, from expression $\rho_b = \eta_{ha}/\rho_a$, the corresponding coefficients of nonuniform heat absorption η_{ha} . With the known values Δi_{mt} and i_{ln} , we can find the unit enthalpy of the working fluid at the exit from a malfunctioning tube, and therefore, its temperature t_{ml}^{ex} . Using these data, we can construct maldistribution characteristics which show variations in the coefficient of hydraulic maldistribution and temperature of the working fluid at the exit from a malfunctioning tube depending on the coefficient of non-uniform heat absorption, i.e. $\rho_h = f(\eta_{ha})$ and $t_{ml}^{ex} = f(\eta_{ha})$, see Fig. 11.20.

The maldistribution characteristics of more heavily heated tubes ($\eta_{ha} > 1$) for ρ_h are falling and for t_{ex} are rising. In a particular portion, curves of ρ_h fall off steeply to a critical state when a slight increase of η_{ha} can lead to a sharp drop in the flow rate and an increase in t_{ex} . This is associated with the fact that in the region of high

heat capacities even a slight increase in the unit enthalpy, which can always occur in some tubes, can cause a sharp increase in the unit volume of fluid, v, and therefore, in hydraulic resistance. This in turn leads to a lower flow rate through the tube, i.e. maldistribution in the tube is aggravated further, while the unit heat absorption Δi increases along with the volume of fluid in it, v. The process continues until a flow rate is established which has a corresponding state of maldistribution of heat at a given heating load. In some cases, this may lead to emergency situations.

The allowable heating load is established for the particular operating conditions of a heating surface. For steam superheaters in which the outlet portions of tubes operate under almost extreme temperature conditions, the allowable heating load should not exceed 15% of the total heat absorption by the superheater. For better reliability, the tube system of steam superheaters is sectionalized in the steam path.

Economizers are mounted in regions of moderate heating rates, and water in them has a relatively low temperature. For economizers, thermal maldistribution of 50% or more is allowable. For this reason, it is not necessary to sectionalize the economizers in the water path.

In evaporating tubes, temperature is relatively low (boiling point). If there are no flow disturbances, and therefore, heat is removed properly, the temperature of the tube walls usually exceeds that of the working

fluid by only 20-30°C. An appreciable maldistribution of heat might he allowed here, but adverse temperature conditions are likely to occur in evaporating tubes, especially at high heating intensities. For this reason, maldistribution of heat in evaporating tubes should be not more than 20-40%.

In supercritical-pressure boilers, the zone of high heat capacities is most sensitive to variations in heat absorption. In this zone, the thermophysical properties of the working fluid vary most significantly with variations in enthalpy. Therefore, it is especially important to construct the characteristic $t_{mj}^{ex} = f(\eta_{ha})$ for this zone.

11.5. Effect of Headers on the Distribution of Working Fluid Between Tubes

There are three main types of headers (Fig. 11.21): (a) supply, or distributing, headers, I, which distribute the working fluid between parallel tubes connected to them; (b) discharge headers, 2, which receive fluid from a number of parallel tubes; and (c) intermediate, or mixing, headers, 3, which are used to equalize non-identity in tube operation.

Mixing headers are most efficient when the flow is single-phase (either steam or water). They are often employed to equalize non-identical operation of tube coils in superheaters.

Supply and discharge headers have different effects on the operation of heating surfaces. In steam superheaters, their effect depends heavily on

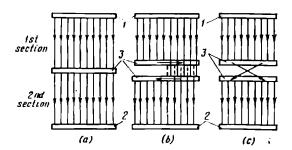


Fig. 11.21. Headers and their connection with the tube system of heating surfaces

⁽a) linear connection; (b) flow transfer through mixing headers; (c) flow transfer through crossover tubes

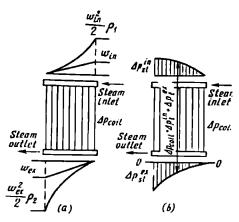


Fig. 11.22. Hydraulic characteristics of a Z-shaped superheater circuit
(a) distribution of velocity and velocity head;
(b) pressure distribution

how steam is supplied to, or removed from, a header. In some plants, headers were used in which steam was admitted and discharged as a concentrated jet through the end faces.

For instance, in a Z-shaped superheater circuit (Fig. 11.22), as steam is distributed between the coils, its axial velocity in the supply header, w_{in} , decreases, and accordingly, the pressure head $w_{in}^2 \rho_1/2$ decreases and changes to static pressure Δp_{st}^{in} (Fig. 11.22a). In contrast to this, the static pressure at the outlet from the discharge header, Δp_{si}^{ex} , decreases. As follows from Fig. 11.22b, the leftmost coils of the superheater operate at a pressure gradient $\Delta p_{coll} + \Delta p_{sl}^{in} +$ $+\Delta p_{st}^{ex}$, which is higher than the pressure gradient Δp_{coll} in the rightmost coils. The difference between the pressure heads in the coils is equal to the sum of static heads in the supply and discharge header, i.e. $\Delta p_{st}^{in} + \Delta p_{st}^{ex}$. The difference in pressure gradients appears not only between the extreme coils, but also between any two sections along the header length and is determined as the sum of the respective values of Δp_{sl}^{tp} and Δp_{sl}^{ex} for these sections.

In other circuits with concentrated supply and removal of steam through

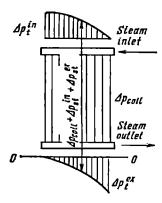


Fig. 11.23. Pressure distribution in a IIshaped hydraulic circuit containing a singlephase fluid

the header end faces, more favourable conditions can form, though the operation of parallel tubes in the system may be substantially non-identical. For instance, with concentrated supply and removal of steam through the header end faces in a II-shaped circuit (Fig. 11.23), the conditions of steam admission are similar to those of a Z-shaped circuit, so that the pattern of static pressure distribution along the supply header remains essentially the same, i.e. pressure increases along the motion of fluid. In the discharge header, static pressure decreases towards its outlet.

As follows from Fig. 11.23, the left-most coils are at a pressure gradient $\Delta p_{st}^{in} + \Delta p_{coil}$ and the rightmost ones at $\Delta p_{coil} + \Delta p_{st}^{ex}$. The difference in the pressure gradients between them is determined by the difference in the static heads in the supply and discharge header, $\Delta p_{st}^{in} - \Delta p_{st}^{ex}$.

Since the unit volume of the fluid in the coils is higher upon heating than before heating, the effect of the discharge headers on fluid distribution between parallel coils in any hydraulic system is more substantial than the effect of supply headers.

The header effect can be decreased either by increasing Δp_{coll} or by decreasing Δp_{sl} in the headers. Both measures are inefficient, however, sin-

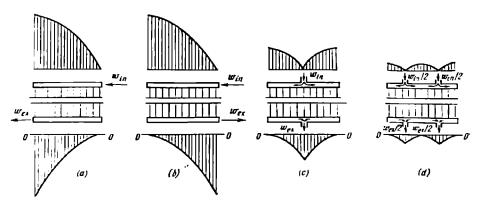


Fig. 11.24. Static pressure distribution along the length of a header depending on the method of supply and discharge of a single-phase flow

ce the former requires a higher pressure in the boiler and a higher auxiliary flow rate, while the latter requires larger headers, and therefore, more metal.

The effect of velocity head can be diminished if steam is introduced and removed not through the end faces of a header (as in Fig. 11.24a and b), but from its middle (Fig. 11.24c). In this case the axial velocity of steam decreases by 50% and the velocity head by 75%. Still better results can be obtained by distributed steam admission to a supply header and distributed removal from a discharge header. With two supply or discharge tubes provided in a header (Fig. 11.24d), the axial velocity and velocity head can be diminished respectively to 1/4 and 1/16 of their values in a header with a single tube. In modern boiler plants, fresh steam superheaters are connected to a number of supply and discharge tubes, so that headers have only a slight effect on steam distribution. In reheaters, where the resistance of tube coils is relatively low and the resistance of headers is high (because of high steam velocities), the effect of headers can be substantial. In economizers, where the unit volume of water is naturally low and the axial velocity in headers inessential, the effect of velocity head presents no problem. In once-through boilers and multiple forced circulation boilers, the resistance of evaporating tube coils is high, so that the effect of pressure variations along the length of headers may be neglected.

11.6. Flow Pulsations

The steady-state operating conditions of a boiler may be disturbed by various factors, leading to pulsations in the working fluid flow. The disturbing factors are variations in the heating intensity, pressure, flow rate and temperature of feed water. There are two main types of flow pulsations: boiler pulsations and intercoil pulsations.

Boiler pulsations are oscillations in the flow rate of the working fluid in individual elements, circuits or the whole plant. They may be caused by sharp variations in the indicated parameters. In similar sections of parallel tubes, flow parameters vary synchronously. Boiler pulsations are attenuating (Fig. 11.25a); they disappear as soon as the disturbance is eliminated.

Flow rate pulsations can attain a level characteristic for the given conditions, after which they do not disappear spontaneously (Fig. 11.25b). This means that the flow rate of water $(\omega \rho)_w$ through some tubes first increases to a maximum, then decrea-

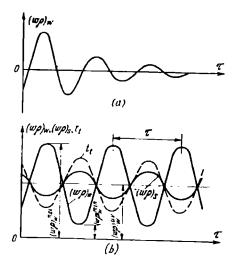


Fig. 11.25. Pattern of attenuating oscillations (a) and self-sustained oscillations (b) in a single tube of a once-through element

ses, passes through the average value. drops to a minimum (sometimes negative), and then increases again. This process may be repeated more than once. In this case, the flow rate of water in other evaporating tubes also pulsates, but the process is shifted in phase. Therefore, the flow rate of water through some tubes increases periodically while at the same time decreasing in other tubes, with the total pressure gradient between the headers of these tubes remaining the same. This phenomenon, called intercoil pulsation, can occur even at a constant total flow rate through the parallel tubes of a section.

The pulsation period in once-through boilers may sometimes be as long as fractions of a minute or a few minutes. With a high amplitude of flow rate pulsations, this may be harmful to evaporating tubes, since heat transfer is impaired during the periods of low flow rate, resulting in temperature variations in the tube walls (curve t_l) and fatigue stresses in the metal. In horizontal tubes, phase separation of the flow may take place, creating the risk of overheating the

upper surface of the tubes and forming fatigue cracks.

Intercoil pulsations can be characterized by the pulsation period τ and amplitude (see Fig. 11.25).

As has been established, intercoil pulsations obey the following regularities:

- (a) they can appear in some tubes of an evaporating section even under steady-state heating and hydraulic conditions in a boiler;
- (b) pulsations of the flow rate of the fluid in parallel tubes of a section are shifted in phase so that the total flow rate and other parameters of the fluid at the outlet from the heating section remain constant;
- (c) the amplitude of flow rate pulsations at the inlet of a tube is much higher than the amplitude at its outlet, while the pulsation period is the same;
- (d) the highest flow rate of water at the inlet corresponds to the lowest steam flow rate at the outlet, i.e. the phases are displaced by 180°;

(e) in intercoil pulsation, the pressure in evaporating tubes oscillates with a period oqual to the period of pulsation of the fluid flow rate.

Since intercoil pulsations are caused primarily by variations in the physical properties of the working fluid in the zone where evaporation begins, the probability of their appearance decreases inversely with pressure. At supercritical pressures, pulsations are less common and have a lower amplitude, but the pattern of flow rate pulsations at subcritical and supercritical pressure is essentially the same.

Flow rate pulsations diminish with an increase in the mass velocity, owing to the resulting increase in hydraulic resistance in the economizer portion of the tubes. The boundary mass velocity in horizontal tubes at which pulsations can occur depends on pressure p, heating load q, length of the heated portion l, and inner diameter of lubes d:

$$(w_{\rm P})_b^h \sim (w_{\rm P})_0 k_p \bar{q} \frac{1}{d}$$
 (11.42)

where $(w\rho)_0$ is the boundary mass velocity at a pressure 10 MPa which can be found on curves plotted for the given conditions of inlet throttling and subcooling, and k_p is a correction factor for the given pressure.

In vertical tubes (coils), the hydrostatic component of the pressure gradient, Δp_h , decreases the probability of the appearance of pulsations, and therefore, decreases $(wp)_h$:

$$(w\rho)_{b}^{v} = c (w\rho)_{b}^{h}$$
 (11.43)

where $(w\rho)_b^b$ is the boundary mass velocity for a similar horizontal tube, [see formula (11.42)] and c is the coefficient for vertical tubes which can be found on curves:

$$c = \frac{(\omega \rho)_b^r}{(\omega \rho)_b^h} = f(\Delta i_{ln}, p)$$

where Δi_{in} is the degree of subcooling of the working fluid at the inlet and p is pressure.

Ascending-descending and slightly inclined tube elements in which Δp_h does not exceed 10% are calculated as are horizontal tubes, with $(w\rho)_b$ increased by 20%. With a higher contribution from the hydrostatic component, the calculation is done as for vertical channels. The actual mass velocity should always be greater than the boundary value: $(w\rho)_w > (w\rho)_b$.

If this condition is not satisfied, the hydraulic resistance of the economizer portion of tubes can be increased by mounting orifice plates. These are arranged at the inlet of the tubes. For once-through boilers, throttling conditions to avoid pulsations should be calculated for the lowest value of mass velocity wp. The degree of thrott ling required to prevent intercoil pulsations also eliminates instability in the hydraulic characteristic of the coils.



HYDRODYNAMICS OF CLOSED HYDRAULIC SYSTEMS

12.1. Laws of Free Circulation

Free-circulation boilers usually have a circulation circuit of evaporating tubes arranged in the boiler furnace. Continuous motion of water and steamwater mixture (circulation) is organized in the circuit, which ensures continuous and efficient removal of heat from the heating surfaces. This makes it possible to maintain the temperature of the tube metal at a tolerable level and thus ensure the reliable and long operation of the circulation circuit.

Free circulation is produced by the driving circulating head S_{dr} which

appears from heating the vertical uptake tubes [see formula (1.1)].

Let us write Bernoulli's equation for each section of the circulation circuit. It is assumed (Fig. 12.1) that a steamwater mixture of density ρ_a moves in section 3-4, and, at the pressure in the drum, water of density ρ' moves in the remaining path, i.e. in sections 4-1-2-3. The reference plane is taken at the header level (2-3).

For section 1-2 (downtake tubes) we have:

$$(H-h) \rho' g + p_1 + \frac{w_1^2}{2} \rho'$$
$$- p_2 + \frac{w_2^2}{2} \rho' + \Delta p_{1-2}$$
 (12.1)

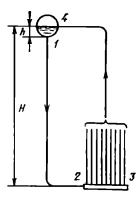


Fig. 12.1. To derivation of the principal equation of circulation

Similarly, for sections 2-3 (lower header), 3-4 (uptake tubes), and 4-1 (water space of the drum) it can be written:

$$p_{2} + \frac{w_{2}^{2}}{2} \rho' = p_{3} + \frac{w_{3}^{2}}{2} \rho' + \Delta p_{2-3}$$

$$(12.2)$$

$$p_{3} + \frac{w_{3}^{2}}{2} \rho' = H \rho_{a} g + p_{4}$$

$$+ \frac{w_{4}^{2}}{2} \rho_{a} + \Delta p_{3-4}$$

$$(12.3)$$

$$H \rho' g + p_{4} + \frac{w_{4}^{2}}{2} \rho_{a} = (H - h) \rho' g + p_{1}$$

$$+ \frac{w_{1}^{2}}{2} \rho' + \Delta p_{4-1}$$

$$(12.4)$$

Adding together the pressure gradients in the sections of the circulation circuit and noting formula (1.1), we obtain:

$$H \left(\rho' - \overline{\rho'_a} \right) g = \Sigma \Delta p_{hr} = S_{dr}$$
(12.5)

With steady-state motion, the pressure difference between the water columns in the downtake tubes and the columns of the steam-water mixture in the uptake tubes is counterbalanced by the sum of hydraulic resistances to the motion of the working fluid in the circuit. Having related all resistances to the downtake and uptake tubes, we get:

$$S_{dr} = \Delta p_{ds} + \Delta p_{des} \qquad (12.6)$$

The difference between the driving circulating head and the resistance of ascending tubes is called the useful circulating head:

$$S_{us} = S_{dr} - \Delta p_{us} \tag{12.7}$$

Comparing formulae (12.6) and (12.7), we find the principal equation of circulation:

$$S_{us} = \Delta p_{des} \tag{12.8}$$

i.e. the useful circulating head is spent to overcome the resistance of the descending sections.

The driving circulating head, and therefore, the useful circulating head depend heavily on the relative velocity of steam (w_r) in the ascending tubes. With the same mass flow rate of the fluid, as the relative velocity of steam increases, the fraction of the tube cross section occupied by steam, q, decreases while that occupied by water, $(1-\varphi)$, increases; thus, the density of the steam-water mixture in the ascending tubes increases. In turn. the relative velocity of steam, and therefore, φ , ρ_m , and S_{us} depend on circulation velocity w_0 , resolved velocity of steam w_0 , pressure p, and tube diameter d:

$$\rho_m = f(w_0, w_0'', p, d)$$

$$S_{u,s} = f(w_0, w_0'', p, d)$$

These relationships are rather complicated and have not yet been solved analytically. Their graphical solution is also impossible in view of the large number of the parameters involved. For this reason, S_{us} is usually represented graphically as a function of w_0 at various values of w_0^* and constant values of the other parameters (p, d). Figure 12.2 shows w_0 - S_{us} curves obtained at various values of resolved steam velocity w_0^* . For the same calculation velocity w_0 , as the heating rate is increased $[(w_0)_2 > (w_0)_1]$, the density of the steam-water mixture in ascending tubes decreases and S_{u_s} increases.

Under identical conditions, the useful circulating head depends on the

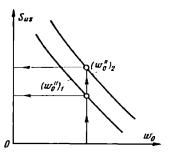


Fig. 12.2. Effect of circulation velocity on the useful head at $w_0'' = \text{constant}$; $(w_0'')_1 < (w_0'')_2$

pressure in the circuit (Fig. 12.3). At a higher pressure, the density of the steam-water mixture in ascending tubes is higher, and therefore, S_{us} is lower. As pressure approaches the critical value, S_{us} decreases to a relatively low value, so that free circulation becomes ineffective. The ultimate pressure at which reliable free circulation is still possible in boilers is equal to 18-19 MPa. On the other hand, even at supercritical pressures. but at low friction losses in the circuit of once-through boilers, free circulation turns out to be sufficient to fire the boiler at a water flow rate that is reduced by 50% and with the recirculation pump switched off.

The effect of relative steam velocity depends on pressure in the circuit. At a low pressure, large formations of steam are possible in a tube (slug flow), and the relative steam velocity

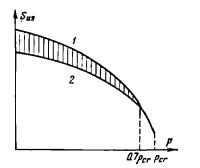


Fig. 12.3. Effect of pressure on the useful circulating head

1-w, neglected; 2-w, considered

is rather high. As the pressure and saturation temperature increase, surface tension decreases and 'slugs' break into finer bubbles, thus decreasing the relative steam velocity. At $p/p_{\rm cr} > 0.7$ (see Fig. 12.3), the effect of relative steam velocity can be neglected.

Let us recall that the actual density of the steam-water mixture ρ_a depends on the actual volume steam content φ [see formula (9.40)]. Using equation (12.5), we obtain a convenient formula for the driving circulating head:

$$S_{dr} = H (\rho' - \overline{\rho}_a) g$$

= $H \varphi (\rho' - \rho'') g$ (12.9)

Formula (12.9) has been derived under the assumption that the ascending tubes of a circulation circuit contain a steam-water mixture along their whole height. Actually, developed boiling begins in the ascending tubes at a certain distance above the inlet and, accordingly, the whole length of the tubes should be divided into an economizer portion H_{ec} and evaporating portion H_{ec} .

The section in which developed boiling begins is called the boiling section (or boiling point). The height of the evaporating portion in ascending tubes can be found by the formula

$$H_{ev} = H - H_{ec} \qquad (12.10)$$

and substituted into formula (12.9) to determine the driving circulating head.

With a non-boiling economizer, the water enthalpy at its outlet, i_{ec} , is less than i', and therefore, the temperature of water in the boiler drum is below the boiling point. The quantity of boiling water that enters the drum is greater than the quantity of feed water by the magnitude determined by the circulation ratio K, so that the degree of water subcooling in the drum can be found as

$$\Delta l_d = \frac{i' - l_{ec}^{\tau}}{K} \tag{12.11}$$

Therefore, in the general case, water at the inlet to the descending tubes is subcooled*. The degree of subcooling increases as water moves in the descending tubes, which is due to an increase in its hydrostatic pressure, and attains a maximum in the lower header of the circulation circuit:

$$\Delta i_{12} = \Delta i_d$$

$$+\frac{\Delta t'}{\Delta p} \rho' g \left(H_{des} - \frac{\Delta p_{des}}{\rho' g}\right)$$
 (12.12)

where $\frac{\Delta t'}{\Delta p}$ $\rho'g$ is the change of water enthalpy per unit height, kJ/(kg m), and Δp_{des} is the hydraulic resistance of the descending tubes, Pa.

This is the degree of subcooling that water has at the entry to the ascending tubes. As it moves there to the boiling section, the hydrostatic pressure decreases and subcooling diminishes by the magnitude:

$$\Delta i_{ec} = H_{ec} \rho' g \frac{\Delta i'}{\Delta p} \qquad (12.13)$$

Therefore, the degree of water subcooling per unit flow rate (1 kg) up to the boiling section is

$$\Delta i_d + \frac{\Delta i'}{\Delta p} \rho' g \left(H_{des} - \frac{\Delta p_{des}}{\rho' g} \right) - H_{ec} \rho' g \frac{\Delta i'}{\Delta p}$$
(12.14)

or for the total flow rate of circulating water, G, kg/h:

$$\left[\Delta i_d + \frac{\Delta i'}{\Delta p} \rho' g \left(H_{des} - \frac{\Delta p_{des}}{\rho' g} \right) - H_{ce} \rho' g \frac{\Delta i'}{\Delta p} \right] G \qquad (12.15)$$

Assuming that heat absorption along the height of the circuit considered is constant, the heat absorbed for the same time in the economizer portion is:

$$Q_{ec} = \frac{H_{ec}Q_c}{H_c} \tag{12.16}$$

where Q_c is the heat absorption of the circuit, $k\mathbf{J}/h$, and H_c is the heated height of the circuit, m.

The height of the economizer portion can be found from the balance of heat for that portion, i.e. the quantity of heat that should be transferred to water per unit time to preheat it to boiling in the economizer portion [formula (12.15)] and the quantity of heat absorbed for the same time by the economizer portion from the furnace [formula (12.16)]. Hence we have:

$$H_{ec} = \frac{\Delta l_d + \frac{\Delta i'}{\Delta p} \rho' g \left(H_{des} - \frac{\Delta p_{des}}{\rho' g} \right)}{\frac{Q_c}{H_c G} + \frac{\Delta i'}{\Delta p} p' g}$$
(12.17)

12.2. Calculation of Circulation Circuits

Free-circulation circuits may be either simple or complex. In a simple circuit (Fig. 12.4a and b), all ascending tubes have the same geometrical characteristics (diameter, length and shape) and are heated under identical conditions. Simple circuits have no common elements with other circuits. An example of such a circuit is the water walls of boiler furnaces. Complex circuits (Fig. 12.4c) may have different geometrical characteristics and different heating conditions. They

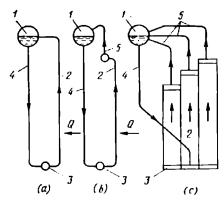


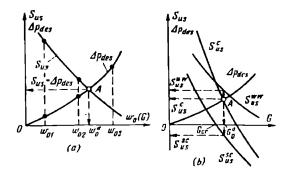
Fig. 12.4. Diagrams of (a, b) simple and (c) complex circulation circuits

1—drum; 2—uptake tubes (water walls); 3—header;
4—downtake tubes; 5—steam-circulating tubes

When all feed water is fed to a bubblecap steam washer (see Sec. 15.4), subcooling is eliminated by the partial condensation of bubbling steam.

Fig. 12.5. Circulation diagrams of simple circuits

(a) with steam-generating tubes connected directly to the drum; (b) with steam-circulating tubes; A-working point; wa-actual circulation velocity (water flow rate)



may have common elements, such as descending tubes which supply the working fluid to the ascending elements of all sections which form a complex circulation circuit [22].

Simple circulation circuits. The principal circulation equation (12.8) is solved graphoanalytically. Both sides of the equation depend on the water flow rate (circulation velocity), i.e. $S_{us} = \int (w_0)$ and $\Delta p_{des} = \int (w_0)$. With an increase in w_0 , the useful circulating head decreases (see Fig. 12.2), while the hydraulic resistance of the descending elements of the circuit increases in proportion to the square of the flow rate. Intersection of the curves S_{u} , and Δp_{des} determines the working point A of the circulation diagram (Fig. 12.5a) whose coordinates satisfy the circulation equation (12.8).

Three different values of circulation velocity w_{01} , w_{02} , and w_{03} are given for constructing the circulation diagram. The calculation determines successively: the hydraulic resistance of the descending tubes in which water flows [see formulae (9.43) and (9.50)]; the respective heights of the economizer portion (12.17) and the evaporating portion (12.10); the relative cross-sectional area occupied by steam (9.34); the driving circulating head (12.9); the resistance of the ascending portion in which the steamwater mixture moves (9.49); and the useful circulation head, (12.7).

The results obtained are used to

construct the circulation diagram (Fig. 12.5a), i.e. the curves $S_{us} = f(w_0)$ and $\Delta p_{des} = f(w_0)$. Then the actual circulation velocity w_0^a is found for the working point of the diagram; together with the quantity of steam formed in the circuit, G_s , it makes it possible to determine the circulation ratio K.

In modern steam boilers, primarily simple circulation circuits are employed in the form of water-wall sections connected to steam-circulating tubes (Fig. 12.4b). In such circuits, the driving head appears both in heated water-wall tubes and unheated steam-circulating tubes, since these are also filled with the steam-water mixture (Fig. 12.5b).

The corresponding useful heads are: S_{us}^{ww} in heated water-wall tubes and S_{us}^{sc} in unheated steam-circulating tubes. The total useful head of the circuit, $S_{us}^c = S_{us}^{ww} + S_{us}^{sc}$, is spent to overcome the hydraulic resistance of the descending tubes Δp_{des} . The coordinates of point A at the intersection of the curves determine the actual water flow rate in the circuit, G_a^a , and useful head S_{us}^c . The actual useful heads of the water-wall and steam-circulating tubes are determined by the ordinates of corresponding points on the curves at G_0^a . The total cross-sectional area of steam-circulating tubes is smaller than that of water-wall tubes, and therefore, at high flow rates $(G > G_{cr})$, their resistance may turn out to be higher than

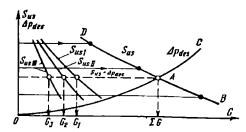


Fig. 12.6. Circulation diagram of a complex circuit

the useful head produced in them, that is, the useful head of water-wall tubes S_{us}^{us} will be partially spent to overcome their resistance.

Complex circulation circuits. For a complex circulation circuit with a common downtake system (Fig. 12.4c), the curves of useful head are constructed in the recommended sequence for each ascending section: $S_{usI} = (S_{us}^{ww} + S_{us}^{sc})_{I}$, $S_{usII} = (S_{us}^{ww} + S_{us}^{sc})_{II}$, $S_{usIII} = (S_{us}^{ww} + S_{us}^{sc})_{II}$ (see Fig. 12.6). Since all the sections of a complex circuit operate in parallel at the same pressure gradient, their circulation characteristics are summed up by adding together the water flow rates at the same values of usoful heads (by the abscissae) to obtain the total characteristic DB. Additionally, a curve of resistance of the downtake (water-feeding) section of the circuit, which is common for all ascending sections, is constructed (curve OC). The intersection of curves DB and OC gives the working point A of the circulation diagram, by which one can find the total water flow rate in the complex circuit, ΣG , and the useful head S_{u_n} .

Water flow rates in the sections are found by drawing the horizontal through the working point up to the intersection with the curves of useful circulating head for each section. When the water flow rate and quantity of steam produced in each section are known, one can determine the actual values of w_0 and K and calculate w_{dee} and the total circulation ratio for the complex circuit.

Nowadays, circulation circuits are calculated by means of electronic computers. This makes it possible, without spending much time, to calculate various variants for various designs and operating parameters of a boiler plant.

Steam generators employed at nuclear power stations with water-cooled water-moderated reactors may be either vertical or horizontal in design (see Sec. 24.2). In both types, the evaporating heating surfaces are formed by tube banks immersed into a large volume of boiling water. The mechanism and hydrodynamics of such systems differ from those of the circulation systems of conventional steam boilers (where boiling water moves in tubes) and have not been yet studied properly. There are no reliable data for the circulation of the driving and useful circulating head nor reliable models of the mechanism of water circulation through tube banks. Furthermore, ascending and descending elements of water flow cannot be distinguished clearly.

On the other hand, water circulation in intertubular spaces is employed widely in the horizontal and vertical steam generators of nuclear power stations in the USSR and other countries. In vertical steam generators, a tube bank is surrounded by a shell, so that it is possible, with certain assumptions, to single out the ascending and descending portion of the circulation circuit. Calculating the circulation circuit is not very difficult and can be made by the standard method of calculating circulation in steam boilers discussed earlier.

In horizontal steam generators, the ascending and descending portions of the circulation circuit are not separated structurally. Circulation takes place in a large volume with a tube bank immersed into it. For this reason, the pattern of water motion through freely immersed tube banks should be analysed before selecting the configuration of the circuit for calculation.

12.3. General Hydraulic Characteristic of Evaporating Tubes and Its Role in Estimating the Reliability of Circulation

The general hydraulic characteristic of an evaporating tube relates the useful circulating head to flow rate and includes the ascending or descending motion of the working fluid. Before constructing the characteristics, let us consider the laws of distribution of density of a steamwater mixture in an evaporating tube as a function of circulation velocity at a constant heating intensity.

As seen in Fig. 12.7, if we neglect the relative velocity of steam w_r ($\varphi = \beta$) and friction in tubes Δp_{fr} , the curve ρ_m will be symmetrical and have a peak in a point corresponding to the zero circulation velocity $w_0 = 0$ and boundary density ρ^n . In other words, with a zero flow rate of water, the tube will be filled only with steam. An increase in the water flow rate at a given heat release results in a denser steam-water mixture which at high values of w_0 is asymptotic to its extreme value ρ^r .

The general hydraulic characteristic of an evaporating tube in a circulation circuit for the given conditions $(w_r = 0 \text{ and } \Delta p_{fr} = 0)$ is a mirror image of the curve ρ_m (curve 1 in Fig. 12.8). For these conditions, $\varphi = \beta$.

As the circulation velocity w_0 decreases, the useful circulating head increases symmetrically and in the limit approaches its maximum:

$$S_{us}^{max} = H(\rho' - \rho'') g$$
 (12.18)

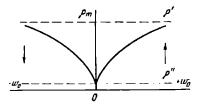


Fig. 12.7. Variations of density ρ_m of steamwater mixture as a function of w_0 in ascending and descending motion

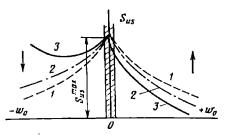


Fig. 12.8. Effect of the steam slippage and friction on the total hydraulic characteristic of a steam-generating tube

1—theoretical circulating head (neglecting steam

1—theoretical circulating head (neglecting steam slippage and friction); 2—steam slippage considered; 3—steam slippage and friction considered

The useful circulating head depends substantially on the relative velocity of steam, this effect being different depending on the direction of motion. In ascending motion, w_r decreases φ and increases ρ_m , resulting in a decrease of S_{us} . In contrast to this, with descending motion, w_r increases φ and decreases ρ_m , and therefore, S_{us} is increased (curve 2). As, however, the circulation velocity approaches zero, the highest useful circulating head will be lower than that found by formula (12.18).

Friction forces are always directed opposite to the motion of fluid, and therefore, decrease S_{us} in ascending tubes and increase it in descending ones (curve 3).

The concept of zero circulation velocity is conditional, since an ascending tube is heated and thus generates steam in an amount corresponding to the absorbed heat. To generate steam, the tube must be fed with water. When there is no circulation through an evaporating tube, water should be supplied at such a rate as to compensate for the quantity of steam generated in the tube; this is called the make-up velocity wmu. The term implies, firstly, that the quantity of water supplied to a tube is low and, socondly, that water can enter the tube from the bottom as well as from the top, i.e. either ascending or descending motion is possible in the tube. The region of very low

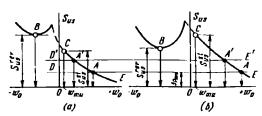


Fig. 12.9. Effect of pressure on the total hydraulic characteristic of a steam-generating tube

(a) lower pressure; (b) higher pressure

positive and negative circulation velocities (hatched area in Fig. 12.8) is usually excluded from analysis.

The right-hand branch of the general hydraulic characteristic corresponds to the ascending motion of fluid, and the left-hand one to the descending (reverse) motion. The relative position of the branches depends on the pressure in the circuit. At a low pressure, the effect of relative steam velocity is high, and therefore, the left-hand branch passes higher than any point on the right-hand branch (Fig. 12.9a). In contrast to this, at a high pressure, the steam velocity and hydraulic resistance are low, and therefore, an appreciable portion of the left-hand branch lies lower than the portion of the right-hand branch in the region of low circulation velocities (Fig. 12.9b). In the right-hand branch, we can find point C which determines the flow rate of water at the make-up velocity, $(\omega \rho)_{mu}$. The ordinate of that point gives us the useful head of stagnation S_{us}^{st} , i.e. the head at which the working fluid virtually ceases to move, resulting in the phenomenon of circulation stagnation. With circulation stagnation, water in a heated tube moves very slowly upwards or downwards, while steam moves only upwards and bubbles through the column of water in the tube. Circulation stagnation may

appear in circuits with evaporating tubes connected to the water space of the drum, i.e. below the water level in the drum (Fig. 12.10a). With tubes connected to the steam space of the drum (Fig. 12.10b), the useful head of water, which moves very slowly, is insufficient to overcome the resistance of the downtake tubes and to raise the working fluid to the topmost level in the uptake tubes of the circuit. Thus, a free water level can form under such conditions. The lefthand branch of the characteristic, which describes steady-state descending motion, has a minimum in point B(see Fig. 12.9). The transition from ascending to descending motion passes through the zero volocity and is called circulation reversal. The ordinate of point B gives us the useful head of reversal, S_{u}^{rev} , i.e. a head at which a change from ascending to descending motion occurs in an evaporating tube.

The distance between the horizontal line DE (Fig. 12.9) and the axis of abscissae is the pressure gradient in downtake tubes $\Delta p_{des} = S_{us}^c$ which, according to equation (12.8), determines the working point Λ in the circulation diagram. As this pressure gradient increases (line D'E' in Fig. 12.9), the water flow rate decreases, but at low pressures the head of stagnation is achieved earlier, and

Fig. 12.10. Connections of evaporating tubes to the drum
(a) Into the water space; (b) into the steam space

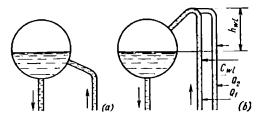
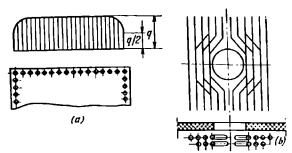


Fig. 12.11. Non-uniform heat absorption by parallel tubes in the water walls of boiler furnaces

(a) across the furnace width; (b) along the height where some tubes are shaded to form a burner port



therefore, circulation stagnation is more probable (Fig. 12.9a); at a high pressure, the head of reversal is achieved earlier and circulation reversal is more likely to occur (Fig. 12.9b).

Circulation circuits are systems of tubes connected in parallel which may be heated differently under actual conditions in a boiler. Non-uniform heating of some tubes may be caused by structural factors in the system or by operating conditions. For instance, tubes in the middle of a furnace wall usually obtain a quantity of heat roughly twice that absorbed by corner tubes (Fig. 12.11a). Tubes around burner ports and at a certain height above the port turn out to be shaded, i.e. they are not directly exposed to the radiant heat of flame (Fig. 12.11b). In the operation of solid fuel-fired boilers, especially when the combustion conditions are disturbed, some water wall tubes may be clinkered by slag across their width or along the height (Fig. 12.12a). Since slag has a low heat conductivi-

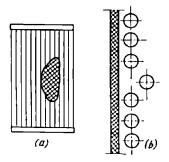
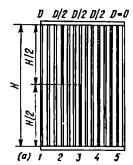


Fig. 12.12. Non-uniform heat absorption by tubes of water walls (a) due to clinkering; (b) due to misalignment of some tubes

ty, its layer will transmit substantially less heat. In some cases, say, if tube fastenings are loosened, some tubes may protrude from a tube row (off-rank tubes), Fig. 12.12b, and will obtain more heat, while adjacent tubes will he shaded by them and receive less heat.

Figure 12.13a shows a system of parallel ascending tubes in a circulation circuit which have identical geometrical and structural characteristics. Suppose that some of them are clinkered. Unclinkered (clean) tubes (say, tube 1) will operate under the rated conditions, i.e. they receive the specified quantity of heat and deliver the specified quantity of steam D. Let tube 2 be clinkered in the top half only so that no steam forms in that portion; the lower half will then absorb half the specified heat, so that the whole tube will produce D/2 of steam. Let tube 3 generate steam in a quantity D/2, but in contrast to tube 2, be clinkered in its lower half only. Let tube 4 be clinkered uniformly along its whole length to such an extent that it delivers steam in the same quantity D/2. Finally, tube 5 is supposed to be clinkered along the whole height to such a degree that it does not receive heat at all, and therefore, produces no steam (D = 0). Assuming that all the tubes are heated uniformly along their height, it may be taken that the quantity of steam in them increases linearly (Fig. 12.13b). As may be seen from the figure, differently heated tubes produce different quantities of steam, and therefore, develop different driving circu-



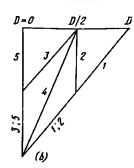


Fig. 12.13. Effect of clinkering of steam-generating tubes on the quantity of generated steam

lating heads; clean tubes deliver the specified quantity of steam and develop the highest driving head: $S_{dr} =$ $= H_s \varphi (\rho' - \rho'')$ g. On the other hand, the clinkered tubes which obtain the same quantity of heat and thus deliver the same quantity of steam D/2, are slagged differently and will develop different driving heads (always lower than that in the clean tubes), since, at the given heat absorption and the corresponding ρ_a , steam fills them to a different height: in tube 3, steam forms only in the top half, H/2; in tube 4, it fills the whole tube height; in tube 2, it also fills the whole tube height, but has a lower density.

Figure 12.14 shows curves of useful circulating heads for a system with non-uniformly heated tubes. As has been demonstrated, the driving circulating heads in the tubes are different. In such a system, all the ascending tubes are connected to com-

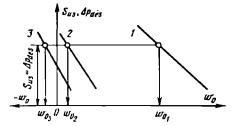


Fig. 12.14. Effect of non-uniform heating of parallel evaporating tubes in a circulation circuit on the direction and velocity of circulation

 $1-S_{us}$ of strongly heated tubes (most tubes in the circuit); $2-S_{us}$ of poorly heated tubes (some tubes in the circuit); $3-S_{us}$ of the worst heated tubes (a few tubes in the circuit)

mon headers and operate under the same forced pressure gradient Δp which is equal to S_{us} [see formula (12.8)]. Hence, the useful circulating head should be the same for all the tubes in the circuit. Owing to non-uniform heating, however, this useful head, which is common for all the tubes, corresponds to different flow rates of the water that circulates in them. With generally favourable temperature conditions for the main number of tubes, uneven heating may result in that little water will pass through some of the tubes. Thus, circulation disturbances, such as stagnation or even circulation reversal, are likely to occur in these tubes.

Circulation circuits are checked for circulation reliability, i.e. for the absence of circulation disturbances, by using the reliability criteria given below. The checking is done for the least heated tubes (with 10% reserve). The criterion of stagnation is:

$$S_{us}^{st}/S_{us}^c > 1.1$$
 (12.19)

the criterion of circulation reversal is

$$S_{us}^{rev}/S_{us}^c > 1.1$$
 (12.20)

and the criterion of free water level is

$$(S_{us}^{st} - \Delta p_{wt})/S_{us}^{c} > 1.1$$
 (12.21)

where Δp_{wl} is the pressure loss for raising the mixture above the water level in the drum, h_{wl} (see Fig. 12.10).

The phenomena discussed are extremely dangerous, since circulation stagnation or the appearance of free water level can interrupt the motion of water in the circulation circuit, while cir-

culation reversal involves a change from ascending to descending motion, i.e. the passage of velocities through zero. All these regimes can disturb heat removal from the internal surface of evaporating tubes and lead to overheating and even burn-out of the tubes.

Circuits operating at pressures p above 11 MPa or with local heat absorption rates q above 400 kW/m² are additionally checked for the probability of heat transfer impairment (burn-out conditions), see Sec. 10.3.

12.4. Hydrodynamics of Descending Tubes and Its Effect on the Reliability of Circulation

Reliable operation of ascending tubes in circulation circuits is ensured by a continuous supply of the required amount of water. A decrease in the water supply through descending tubes to intensively heated ascending tubes can lead to insufficient cooling and, furthermore, to overheating of the tubes. The situation is dangerous and most often results in burn-out of the heated evaporating tubes. If water supply is fully stopped, tho temperature of the walls of heated tubes may rise at a rate of up to 20-25° C/s, meaning that 10-15 seconds are sufficient to get the plant out of order.

Descending tubes of a circuit may turn out to have a water flow rate too low to ensure proper temperature conditions in evaporating tubes if they have an excessive hydraulic resistance or if steam is entrained from the drum into them.

Hydraulic resistance of descending tubes. When analysing the operation of descending tubes, we should distinguish between two cases: (a) water supplied from the economizer into the boiler drum at the saturation temperature (boiling economizer, $t_{cc}^{re} = t'$) and (b) water fed at a temperature below the saturation point (non-boiling economizer, $t_{cc}^{re} < t'$).

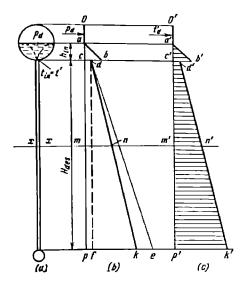


Fig. 12.15. Distribution of pressures and the corresponding enthalpies of water at the saturation line along the height of downtake

With a boiling economizer, water temperature at the inlet to the descending tubes is $t_{in} = t'$ (Fig. 12.15a). As water moves through the drum to the inlet of the descending tubes, its velocity is negligible, so that the velocity head due to the non-parallel drum walls can be neglected. Pressure changes can only occur due to a rise in the hydrostatic pressure by a linear law (line ab in Fig. 12.15b). The pressure at the inlet to the descending tubes is higher than that at the water level in the drum by the magnitude (section bc):

$$\Delta p_d^h = h_{in} \rho' g \qquad (12.22)$$

Upon entry into descending tubes, water velocity increases substantially, since the hydrostatic pressure decreases because of the appearance of velocity head and inevitable energy losses on local resistances. Thus, the pressure at the inlet to the descending tubes decreases (section bd) by:

$$\Delta p_{in} = (1 + \xi_{in}) \frac{w_{des}^2}{2} \rho'$$
 (12.23)

With further motion of water in the descending tubes, pressure again increases, due to an increase in the hydrostatic pressure of the water column along line de by the length of section ef:

$$\Delta p_{des}^h = H_{des} \rho' g \qquad (12.24)$$

or, if we additionally consider the pressure loss by friction and on local resistances

$$\Delta p_{des} = \left[H_{des} - \frac{w_{des}^2}{2g} \right] \times \left(\frac{\lambda t}{d} + \Sigma \xi \right) \rho' g \quad (12.25)$$

where

$$\frac{w_{des}^2}{2g}\left(\frac{\lambda l}{d} + \Sigma \xi\right) < H_{des}$$

Referring to Fig. 12.15, as water moves in the descending tubes, its pressure increases continuously, except for the tube inlet proper, where it decreases somewhat. If, however, assume that $\Delta p_d^h > \Delta p_{in}$, i.e.

$$h_{in} > (1 + \xi_{in}) \frac{w_{des}^2}{2g}$$
 (12.26)

it will follow from Fig. 12.15b that the water pressure in any point of the descending system exceeds its pressure at the water level in the drum p_d . The enthalpy of boiling water is distributed according the increasing pressure (a'b'd'k')Fig. 12.15c). The actual enthalpy of water in any section of the descending tubes is equal to i_d ; it corresponds to the pressure at the water level in the drum p_d and is constant if there is no heating of the descending tubes (line a'p'). The horizontal sections between these lines determine the degree of water subcooling in the respective sections of the descending tubes; for instance, the subcooling in section x-x is determined by the section m'-n'. Therefore, if condition (12.26) is observed, boiling and steam generation in unheated descending tubes is impossible.

If the pressure drop at the inlet to the descending tubes Δp_{in} is equal

to or greater than Δp_d^h , the pressure of water at the inlet to the descending tubes will drop to such a level that the water will turn out to be superheated at the drum pressure or will boil in the zone of reduced pressure. Steam bubbles thus formed will be entrained by circulating water into the descending tubes where the pressure is higher; thus, water will turn out to be subcooled there and steam bubbles will condense. Complete condensation of steam requires a certain time. As a result, steam bubbles will be carried by the water flow an appreciable distance in the descending tubes, i.e. these will be filled with a steam-water mixture, rather than with water, at a substantial length, which will noticeably increase the resistance of the descending tubes and somewhat diminish the driving circulating head. The presence of a slight quantity of steam in the descending tubes is, however, not dangerous.

Appearance of steam in descending tubes. The main causes for the appearance of steam in descending tubes* are as follows:

- (1) formation of steam cones at the entry to the descending tubes, i.e. the direct entrainment of steam through the tubes from the steam space of the drum, and
- (2) entrainment of steam bubbles from the water space of the drum by circulating water.

Formation of steam cones. The operation of evaporating tubes becomes less reliable if steam forms in the zone of reduced pressure at the entry to the descending tubes, i.e. when condition (12.26) is not observed. In such a case, steam cones may form in the water surface in the drum (see Fig. 12.15a), so that large quantities of steam will pass from the steam space into the system of descending tubes and thus disturb water circulation in the circuit. The formation of steam cones is possible in boilers with

Steam can also form in descending tubes on a sudden pressure drop in the boiler.

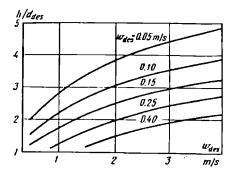


Fig. 12.16. Determination of the minimal water level above the inlet to downtake tubes

both free and forced circulation. Their formation can be prevented by maintaining the water level in the drum at a safe height above the inlets to the descending tubes. The probability of formation of steam cones increases with an increase in the diameter of the descending tubes. As recommended by the standard method of hydraulic calculation of steam boilers [22], the ratio of the minimal water level h to the diameter of descending tubes d_{des} should be chosen by the curves given in Fig. 12.16.

With a non-boiling economizer, the temperature of feed water supplied to the drum is below the saturation point. This water is mixed with boiler water and forms a mixture whose temperature is below the saturation point under pressure at the water level in the drum, $t_{in} < t'$. Water at this temperature enters the des-

cending tubes. This degree of subcooling is equivalent to the degree of subcooling provided by forming a proper water level in the drum.

Water subcooling in the drum is not a useful means of ensuring the stable operation of descending tubes in view of the complexity and inconvenience of this method. As with a boiling economizer $(t_{ln} = t')$, the stable operation of descending tubes in a circuit with non-boiling economizer $(t_{ln} < t')$ is achieved by preventing the formation of steam cones above the entry to the descending tubes. Subcooling of water at the inlet to descending tubes at $t_{ln} < t'$ merely ensures a certain reserve of reliability.

Entrainment of steam from the water space of drum. The normal supply of water to the descending tubes can be disturbed if steam bubbles in the water space of the drum are entrained by water and carried into the descending tubes. When velocity of water flows in the drum is substantial. steam bubbles below the water surface have no time to separate and are thus entrained by the water into the descending tubes. Such conditions may appear, for instance, when the ends of the evaporating tubes in the drum are arranged too close to the inlets of the descending tubes. The entrainment of steam into the descending tubes is prevented by properly organizing the water flows, for instance, by arranging partitions or by separating the steam-water mixture in cyclones.



HYDRODYNAMICS OF BUBBLING SYSTEMS

13.1. Laws of Bubbling

In free-circulation boilers, the working fluid moves as it is heated in ascending tubes; in once-through and

multiple forced circulation boilers, its motion is caused by the pressure head developed by the feed pump. In either case, the fluid in evaporating tubes is two-phase and its motion is

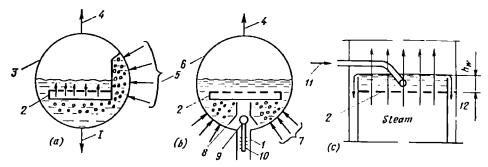


Fig. 13.1. Devices for steam bubbling through water layer

(a) In boiler drum; (b) In separating drum of channel-type; boiling reactor; (c) in stam washer; 1—downtake tubes; 2—distribution plate; 3—drum; 4—steam-circulating tubes; 5—steam-generating tubes; 6—separating drum; 7—steam-water mixture from process channels; 6—steam box; 9—feed water header; 10—distributing perforated tubes; 11—feed water (wash water); 12—wash water drainage

essentially the combined motion of the two components, water and steam, which may have different velocities. Three principal cases are then possible:

 $w_{\bullet} > 0$, $w_{\omega} > 0$ —ascending motion; $w_{\bullet} > 0$, $w_{\omega} < 0$ —flow reversal, and $w_{\bullet} < 0$, $w_{\omega} < 0$ —descending motion.

In contrast to these cases, in which both phases are in motion, there may be a special case when only the lighter phase (steam) is moving, while the heavier phase (water) is stagnant, i.e. it has a zero average velocity: $w_s > 0$, $\bar{w}_w = 0$. This is known as the process of steam bubbling through liquid. Steam bubbling is a specific kind of motion of a two-phase mixture in which bubbles of the lighter phase (steam) rise through the bulk of the

heavier phase (water). Bubbling occurs in the drums of boilers and steam generators when the steam-water mixture is introduced below the water level, in the separating drums of channel-type boiling reactors, in steam washers, etc. (Fig. 13.1). It can also occur in evaporating tubes on certain hydrodynamic disturbances of flow, such as the formation of a free water level or circulation stagnation (see Sec. 12.3). The layer of the steam-water mixture in which steam bubbling takes place is called the dynamic (movable) two-phase layer.

Bubbling is usually effected by feeding steam under a perforated distribution plate (Fig. 13.2). As steam moves through holes in the plate, its jets break into separate bubbles which rise through the bulk of water above the plate to the

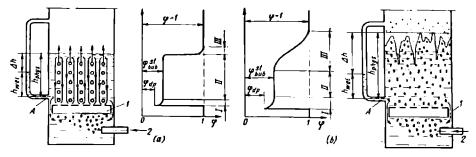


Fig. 13.2 Analysis of the bubbling process and steam content distribution along the height of apparatus

(a) at low w_0^* ; (b) at high w_0^* ; I-III—zones of dynamic two-phase layer; I—steam cushion; 2—nozzle

separating surface between the phases (which is usually called the disengagement surface). The disengagement surface is not smooth but seething, or turbulent, with high splashes being formed continuously by steam bubbles which rise to the surface. In the course of bubbling, steam bubbles entrain water which then moves downward at the walls and in the spaces between bubble chains and is thus forced into circulation. As a result, a zero average flow rate of water $(\bar{w}_w = 0)$ and a positive flow rate of steam $(w_o > 0)$ are established in bubbling.

Bubbling is effected in a bubbling apparatus, such as a vertical column (Fig. 13.2). The water level indicated by the water-level gauge glass at the column is lower than that in the column proper, since water in the apparatus is at the boiling point and, besides, is saturated with steam bubbles and forms a steam-water mixture of a density ρ_{bub} . Water in the gauge glass is subcooled below the saturation temperature at the pressure of bubbling and its density is ρ_g . Since the pressure at the bottom of two communicating vessels (point should be the same, we have:

$$h_{ph}\rho_{bub}g = (h_{wel}\rho_g + \Delta h\rho_s) g (13.1)$$

Noting that the density of steam ρ_s is much lower than that of water in the gauge glass, ρ_g , we obtain:

$$h_{ph}\rho_{bub} = h_{wel}\rho_g \qquad (13.2)$$

Since ρ_{bub} is smaller than ρ_g , the physical level h_{ph} established in the column is higher than the weight level h_{wei} observed in the gauge glass by the magnitude Δh . The physical level in the apparatus is essentially the level of the working fluid at a steam content φ close to unity. The difference Δh is called the water swell.

The supplied steam is distributed over the column cross section according to the hydraulic resistance of the water layer; with a concentrated steam supply, this resistance may be different in various points of the cross section. A uniform distribution of steam can be attained by mounting a perforated steam-distribution plate in the water space. The hydraulic resistance of the plate is much higher than that of the free cross section of the column and is thus a decisive factor in steam distribution. The plate is perforated evenly, and therefore, steam is distributed uniformly over the cross section. A gap is left between the plate and column walls to permit return flow of the water. The edges of the plate are bent downward to prevent a sudden outburst of steam at the walls and to form a steam *cushion* beneath the plate.

A steam cushion is essential for normal operation of the steam-distribution plate in which steam bubbles flow continuously through its holes. At the moment of formation of a steam cushion, steam has a definite (minimal) velocity w_{\min} in the perforations of the plate. For the stable existence of the steam cushion, the actual velocity of steam flow through the perforations must be higher than tho minimal value, i.e. $w'' > w_{\min}$. This is easily accomplished if perforations have a diameter $d_1 = 2-3$ mm, i.e. smaller than the break-off diameter of steam bubbles.

In the steam boilers of thermal power stations and the steam generators of nuclear power stations, submerged distribution plates have larger holes of a diameter of 8-12 mm or even more. Under such conditions, a steam cushion can form if steam passes through the holes in continuous jets. Steam jets can entrain a slight quantity of water droplets from the water space of the apparatus, but this has no substantial effect on the hydrodynamics of the distribution plate.

The hydrostatic head needed to form a stable steam cushion is determined by the difference in the masses of water and steam columns of a height equal to the thickness of cushion δ:

$$\Delta p = \delta \left(\rho' - \rho'' \right) g \qquad (13.3)$$

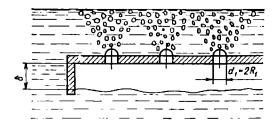


Fig. 13.3. Analysis of operation of a submerged distribution plate

This head is spent to overcome the hydraulic resistance of perforations and to form an excessive pressure required to break through the water film at the exit of steam into the water space above the plate (Fig. 13.3). A water film can be broken through by a force $2\pi R_1 \sigma$ which develops a pressure $2\pi R_1 \sigma / \pi R_1^2 = 2\sigma / R_1$. Then we have:

$$\Delta p = \xi_{perf} \frac{(w'')^2}{2} \rho'' + \frac{2\sigma}{R_1}$$
 (13.4)

The combined solution of equations (13.3) and (13.4) gives the minimal thickness of a stable steam cushion:

$$\begin{split} \delta_{\min} &= \frac{2\sigma}{R_1 (\rho' - \rho'') g} \\ &+ \xi_{perf} \frac{\rho'' (\omega'')^2}{2g (\rho' - \rho'')} \end{split} \tag{13.5}$$

where σ is the surface tension, ξ_{perf} is the coefficient of resistance of the steam-distribution plate which is determined by the free cross-sectional area of its perforations; $R_1 = d_1/2$.

A steam-distribution plate with perforation holes can operate efficiently, with a stable cushion, only in a narrow range of heating loads near the rated boiler load. With an increase in the steam-generating load, the height of the steam cushion increases in proportion to the square of the ratio of loads $(D_x/D_r)^2$, and may result in the break-through of steam at the plate edges. At lower loads, steam is distributed unevenly over the cross section.

A new type of steam-distribution plate has been developed at the Krzhizhanovsky power engineering institute: it has numerous pipes perforated along their whole length and closed at the bottom ends, which are mounted in the holes of a distribution plate (Fig. 13.4). The number and arrangement of pipes and the number of perforations in them are determined by the desired range of variation of the steam-generating load.

Steam accumulates in the water space under the plate and forms a steam cushion of a height up to the length of the perforated pipes. It flows through their perforations into the water layer above the distribution plate. At a change in the steam-generating load, the number of 'active' perforations is self-regulated by a change in the height of the steam cushion. This ensures uniform bubbling and uniform load on the disengagement surface in a wide range of steam-generating loads.

As in forced motion, the principal characteristic of bubbling is the relative cross-sectional area occupied by steam, φ_{bub} . The distribution of steam and water along the height of a bubbling apparatus is different, and therefore, φ_{bub} can vary within a wide

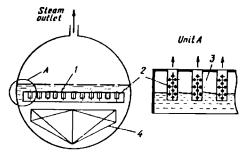


Fig. 13.4. Steam distributor

1—perforated steam-distribution plate; 2—perforated tubes; 3—steam cushion; 4—steam-generating heating surface

interval: from the value corresponding to the steam flow rate immediately above the distribution plate up to almost unity in the steam cushion and in the steam space. The distribution of φ_{bub} along the height depends mainly on the steam flow rate or, what is the same, on the resolved steam velocity w_0^* .

For a particular value of w_0 , one can distinguish between three zones in the dynamic two-phase layer above the distribution plate (Fig. 13.2). With an amplo steam supply, this layer is filled with steam, and therefore, $\varphi_{bub} = 1$. The first zone of the two-phase dynamic layer is immediately above the plate. In this zone, the steam flow is stabilized: the larger bubbles break down into finer bubbles, while fine bubbles combine to form larger bubbles, i.e. all bubbles are essentially transformed into the same stable size. The steam content φ_{bub} changes from the value corresponding to the relative free cross-sectional area of the plate φ_{pl} to a certain constant value φ_{bub} which is determined by the particular conditions of bubbling: steam flow rate and pressure. The initial zone of the dynamic layer has a limited height, usually of a few centimetres.

The second zone is characterized by a constant value $\varphi_{bub}^{t} = \text{constant}$ which has been attained due to stabilization at the exit from the first zone. The bubbling process in the stabilized zone is described most accurately by the generalized formula proposed by M. A. Styrikovich and S. S. Kutateladze [31]:

$$\varphi_{bub}^{st} \approx 0.4 \left(\frac{\rho^*}{\rho'}\right)^{0.15} \left(w_0^*\right)^{4} \frac{\overline{\rho' - \rho''}}{\sigma_g}^{13.6}$$

The formula is applicable at steam content $\varphi_{bub}^{st} \leq 0.7$.

In the third zone (transient zone), φ_{bub} increases continuously from the stabilized value φ_{bub}^{rl} to $\varphi = 1$, which exists above the two-phase dynamic layer. The height of the third zone

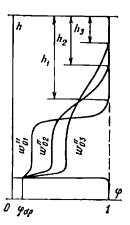


Fig. 13.5. Effect of resolved steam velocity on the distribution of φ along the height of apparatus $w_{01}^{*} < w_{02}^{*} < w_{03}^{*}$; h_1 , h_2 , h_3 —respective heights of steam space

depends on \boldsymbol{w}_{a}^{r} . At a low \boldsymbol{w}_{a}^{r} , individual rising bubbles are distributed in a relatively large water volume and therefore cannot influence one another. Rising bubbles only slightly deform the disengagement surface, so that there is a distinct boundary between the second and third zones and the latter has a low height (curve w_0) in Fig. 13.5). With an increase of w_a^* , the quantity of bubbling steam increases, bubble chains move in the water and combine into steam jets. and a return circulation of water appears in the apparatus. The number of steam jets then increases and impedes the return circulation of water, i.e. water entrained by the rising steam cannot flow back as easily and is retained for a longer time in the upper portions of the dynamic layer, thus leading to water swell. The phaseseparating surface becomes less distinct, the transition zone of the dynamic layer increases in height, and the height h of the steam space in the apparatus diminishes accordingly (curve w_{a}^* in Fig. 13.5). The hydrodynamics of bubbling depends substantially on pressure. With an increase in pressure, steam density increases, so that steam bubbles rise in water more slowly and are retained in the water space for a longer time. This increases φ_{bub}, and therefore, causes water swell and an increase in the height of the stabilized and transient zones. Thus, an increase in pressure is qualitatively equivalent to an increase in w_a^r . For a given weight level, the height of the transient zone, h_{tr} , depends on the fraction of the crosssectional area occupied by steam. The height of the transient zone has a direct effect on the moisture content of the steam supplied from the apparatus into a steam separator. With a higher h_{tr} , the height of the steam space is lower (see Fig. 13.5) and the moisture content of the steam is higher.

13.2. Dynamic Layer in Steam Washers

The washing of steam by pure water is a widely employed method for improving steam quality (see Sec. 15.4). It is commonly effected by passing the steam through a water layer that is retained on a perforated plate. Water overflows the peripheral enclosure of the plate, whose height is chosen so as to obtain the desired water level (Fig. 13.1c), but cannot flow through the holes, since the water column above a hole is acted upon by a force $F_{\Delta p}$ which appears due to the difference of steam pressure below the plate and above the water level. This force is equal to the hydraulic resistance for the passage of steam through the perforated holes and through the water layer above them:

$$F_{\Delta p} = \xi_{perf} \frac{(w'')^2}{2} \rho'' \frac{\pi d^2}{4}$$

$$+\xi_w \frac{(w'')^2}{2} \rho'' \frac{H}{d} \frac{\pi d^2}{4}$$
 (13.7)

Up to the moment the steam breaks through a hole, this force is counterbalanced by the gravity force of the mass of the water column:

$$F_{e} = \frac{\pi d^2}{4} g \rho' h_w \qquad (13.8)$$

and by the force due to surface ten-

$$F_{\sigma} = \pi \ d\sigma \tag{13.9}$$

that is

$$F_{AB} = F_{a} + F_{a} \qquad (13.10)$$

Substituting from formulae (13.7) to (13.9) into (13.10), we obtain the formula for steam velocity w" in the holes of a perforated plate, that enables the water layer to be retained on the plate:

$$w'' = \sqrt{\frac{2g\rho'}{\xi\rho''} \left(h_w + \frac{4\sigma}{\rho'gd}\right)} \quad (13.11)$$

in which \$\xi\$ is the resolved coefficient of resistance of the bubbling device:

$$\xi = \xi_{perf} + \xi_w \frac{H}{d} \qquad (13.12)$$

In formulae (13.7) to (13.11), d is the diameter of holes in a perforated plate, h_w is the level of water above the plate, H is the actual height of the bubbling layer, and ξ_{perf} and ξ_w are the coefficients of resistance to the steam flow in a hole and in the wash water layer.

The height of the steam-washing layer of water is usually not high (50-70 mm) and its resistance is low compared with that of perforated holes, i.e. $\xi_w < \xi_{perf}$. Therefore, it may be assumed in formula (13.11) that $\xi = \xi_{perf}$.

that $\xi = \xi_{perf}$. Formula (13.11) gives a certain reserve of the velocity of steam flow through perforated holes since the actual velocity of steam is higher and is certainly sufficient to ensure the hydrodynamic stability of steam washing by bubbling.

13.3. Effect of Non-uniform Heat Release and Impurities on the Dynamic Two-phase Layer

Boiler water may contain surfaceactive substances which are concentrated mainly at the boundaries between phases. This increases the strength of water films which envelop and diminish steam bubbles. Finer

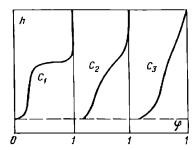


Fig. 13.6. Distribution of φ along the height of apparatus at different concentrations $(c_1 < c_2 < c_3)$ and constant weight level

bubbles rise more slowly in water. Stronger water films are broken with a certain time delay, so that the process of passage of steam bubbles into the steam space is retarded. Under such conditions, the dynamic two-phase layer is saturated with a greater amount of steam, thus leading to water swell and an increase in the height of the layer.

The general distribution pattern of steam content along the height of the dynamic two-phase layer at a constant steam flow rate and various concentrations of impurities in water is shown in Fig. 13.6. As may be seen, the steam content of at a given concentration of impurities increases along the layer height. At a low concentration of impurities the steam content at the exit from a steamdistribution plate is not high. It remains almost constant to an appreciable height (op increases slowly), increases rapidly in the transient zone, which has a low height, and attains a value $\varphi \approx 1$ above the physical water level. With an increase in the concentration of impurities in water, the steam content at the exit from the plate increases and the length of the zone where ϕ is almost stabilized becomes smaller (or even completely disappears at a high concentration). In contrast to this, the height of the transient zone increases substantially.

In steam-generating plants, the generation of steam may be substantially non-uniform across the cross-sec-

tion of an apparatus. This is mainly due to a non-uniform heating intensity which, for instance, in the high-capacity horizontal-type steam generators of nuclear power stations may vary along the length of evaporating elements by a factor of two or three (Fig. 13.7).

The heating intensity can be equalized either naturally, under the effect of a water layer for steam bubbling, or by means of various devices, such as a submerged steam-distribution

plate.

Figure 13.8 shows the curves of steam content φ at various levels along the height of a bubbling layer with a symmetrical (steam is supplied at the centre of the cross section) or asymmetrical initial non-uniformity in a vertical column without a perforated plate, where steam content is equalized naturally. As may be seen, the steam content at the entry to the two-phase layer is distinctly non-uniform, but is equalized along the height due to bubbling in a free volume of water.

A non-uniform steam content leads to the appearance of transverse gradients of density in the steam-water mixture that fills in the apparatus. In turn, these gradients lead to oriented convective currents which equalize the steam content and density. Figure 13.8 shows curves of ϕ for four sections at various heights in the bubbling layer. As may be seen, non-uniformity decreases along the height due to steam redistribution, though complete equalization is not achieved

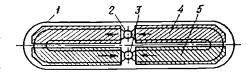


Fig. 13.7. Zone of different steam-generating intensity in the steam generator of a water-cooled water-moderated power reactor (horizontal section)

1—housing; 2—coolant in; 3—coolant out; 4 zone of intensive steam generation; 5—zone of moderate steam generation

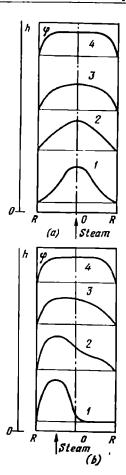


Fig. 13.8. Curves of steam content at various levels (1, 2, 3, 4) along the height of a layer with (a) symmetrical and (b) asymmetrical initial non-uniformity

since the water entrained by bubbling steam flows down mainly along the walls. In columns of a large diameter, the effect of walls is smaller, so that ascending and descending currents, and therefore, the steam content, are distributed more evenly over the cross section.

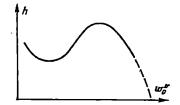


Fig. 13.9. Effect of resolved steam velocity w_0^* on the height at which φ is equalized

The effect of w_0 on the height at which flow equalization is achieved is shown clearly in Fig. 13.9. The left-hand portion of the curve relates to the low values of w_0^* , and consequently, low density gradients, so that flow equalization occurs at a greater height. At a higher steam-generating intensity, flow equalization occurs more quickly, notwithstanding the higher initial non-uniformity. Therefore, the height of stabilization is lower. It might be expected that this trend would continue with a further increase in w_0^* which leads to an increase in the density gradient. Actually, however, at very high flow rates of steam, the axial component of the velocity of the steam-water mixture increases sharply and the process is extended along a greater height in the layer. In a certain range of high values of w_0 , the effect of velocity is predominant and the height of the equalization layer continues to increase. Finally, at very high flow rates of steam (w_0) , it begins to carry off much moisture from the layer. Extrapolation of the curve to its intersection with the axis of abscissae will give a point at which water will be completely carried off from the apparatus, i.e. the process of steam bubbling will change to the forced motion of the steam-water mixture.



PHYSICO-CHEMICAL PRINCIPLES OF BEHAVIOUR OF IMPURITIES IN WORKING FLUID

14.1. Impurities in Feed Water and Their Effect on Equipment

During the operation of any type of boiler-turbine installation the working fluid becomes contaminated with impurities. The quantity and composition of impurities depend on the type of plant, composition of structural materials, and operating conditions. The principal sources and compositions of impurities in the aqueous

heat carrier of thermal power stations are given in Table 14.1.

Impurities may be present in boiler water in either a dissolved or a suspended state. Under particular conditions, they can precipitate from water and form deposits on the heating surfaces, thus impairing heat transfer and raising the temperature of tube walls. Deposits are especially dangerous in intensive heating zones (water walls in boiler furnaces and fuel elements

Table 14.1. Principal Sources and Compositions of Impurities in an Aqueous Heat Carrier

Sources	Principal Impurities
Inleakages: in condensers	Salts (chlorides, sulphates and bicarbonates of calcium, magnesium and sodium), colloidal impurities (organic matter, silicic acid), suspended matter and gases (O ₂ , CO ₂ , N ₂)
in feed water and tap water heaters	Salts (calcium, magnesium and sodium chlorides, sulphates and bicarbonates), silicic acid, and gases
Make-up water (demineralized, distilled)	Sodium compounds, products of metal corrosion lized water), CO ₂ (in distillate)
Softened water	Sodium compounds, silicic acid, gases (the composition of gaseous impurities depends on the water treatment method)
Products of corrosion of structural materials	Oxides of Fe, Cu, Cr, Ni, Zn, Co, Al, etc.
Products of radiolysis and other processes under the effect of neutron fluxes	Radioactive products of corrosion of metals: Fe, Mn, Co, Al, Zr, etc.; gases: N2, O2, Xe, Kr, etc.
Water additives	Phosphoric salts, ammonia, hydrazine, chelates

in nuclear reactors). At nuclear power stations, radioactive deposits can characterize the radiation situation of the equipment.

Impurities can partially pass from water to steam and form deposits in superheaters and in the steam path of turbines. Deposits in superheaters are intolerable since the outlet portions of their coils even at rated heating loads operate at the upper admissible temperature limit of the tube metal. Evon a slight layer of deposits can then raise the metal temperature to an inadmissible level and promote creep phenomena and scale formation.

Deposits in the steam path of turbine are also extremely undesirable. They increase the roughness of the blades and friction losses and therefore diminish turbine efficiency. Heavy deposits in the steam path of turbine can cause additional axial pressure, requiring a decrease in the turbine power. The effect of deposits is especially perceptible in high-pressure turbines where the unit volume of steam is lower and the high-pressure section is accordingly smaller in size.

Methods have been developed to minimize the passage of impurities into water with inleakages in condensers and with make-up water. It is much more difficult to provent water contamination with the products of corrosion of structural materials, especially in plants operating at nearly critical or supercritical pressures. In operation, food water is allowed to have a certain composition and concentration of impurities depending on the type of plant and its water balanco: hundredths of a milligram per kilogram for once-through boilers and a few tens of milligrams per kilogram for drum-type boilers.

14.2. Solubility of Impurities in an Aqueous Heat-transfer Agent and Formation of Deposits

In a sufficiently wide range of high and supercritical parameters of a homogeneous aqueous heat-transfer agent (steam and water), the thermodynamic solubility of low-volatile inorganic substances as a function of the properties of the solvent is determined by two parameters: density and temperature. Solubility can be described by the equation proposed by Prof. O. I. Martynova:

$$\ln C = m \ln \rho - \frac{\Delta H}{RT} + \frac{\Delta S_y^2}{R} \quad (14.1)$$

where C is the solubility of a substance in an aqueous heat-transfer agent, ρ is the density of H_2O at the given parameters of the process, ΔH is the thermal effect of dissolution, R is the universal gas constant, T is the temperature, ΔS is the entropy of dissolution of the substance, and m is the coordination number (hydration characteristic).

Calculations of the solubility of substances in an aqueous heat-transfer agent by formula (14.1) are possible if there are reliable data on the three parameters which characterize the dissolution process: ΔH , ΔS , and m. These data are only available for certain impurities and for a limited range of parameters. For this reason, laws of dissolution of various impurities in water and steam are mainly studied experimentally. The general laws of dissolution will be discussed below.

Dissolution of impurities in water and laws of formation of deposits. All substances present in boiler water can be divided into two groups: slightly soluble and readily soluble. The former group includes calcium and magnesium salts and hydroxides and oxides of structural materials with which the aqueous heat-transfer agent may have contact. Solubility curves of selected slightly soluble impurities in hot water are shown in Figs. 14.1 through 14.3. The group of readily soluble impurities which are found in the water of steam-turbine plants includes sodium salts and sodium hydroxide. Their solubility curves are shown in Fig. 14.4. As soon from the figures. the solubility of some substances

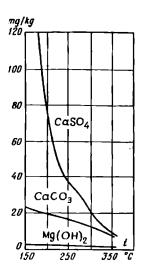
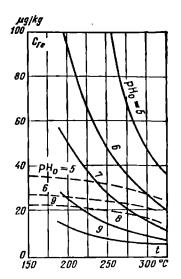


Fig. 14.1. Solubility of principal scale-formers in water



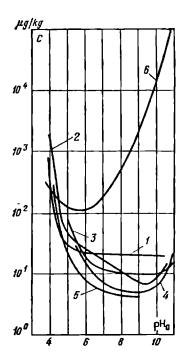


Fig. 14.3. Solubility of metal oxides in boiling water (p = 7 MPa)

1—Pe (oxidizing medium): 5—Fr (reducing medium); 5—Cu; 4—Zn; 5—Ni; 6—Al

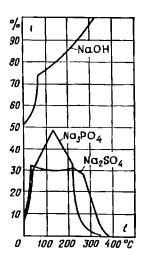


Fig. 14.4. Solubility of readily soluble compounds in water

increases with temperature, while that of others decreases.

Slightly soluble impurities. Calcium and magnesium compounds and metal oxides which belong to the group of slightly soluble impurities, come into the steam-water path of steam-turbine plants from different sources and, what is most important, behave differently in an aqueous heat-transfer agent. For this reason, they will be discussed separately.

In normal operation of boilers, the concentrations of slightly soluble impurities in feed water, mainly of calcium and magnesium salts, are low. They may increase on certain disturbances in the system of condensate cleaning and water treatment or due to excessive inleakages in condensers.

In any electrolytic solution, a dissolved substance partly dissociates into ions (cations Me^{m+} and anions Acⁿ⁻) and partly remains in the form of molecules. The degree of dissociation, i.e. the fraction of dissociated molecules, depends on the properties of the solute and temperature.

For a saturated solution at a given temperature, the product of active concentrations of ions $a_{\text{Me}^{m+}}^n \cdot a_{\text{Ac}^n}^m$ which is called the solubility product, is constant, i.e.

$$SP = a_{Me^{m+}}^n \cdot a_{Ae^{m-}}^m \qquad (14.2)$$

where n and m are stoichiometric coefficients.

For substances with a positive temperature coefficient of solubility, d(SP)/dt > 0 and for those having a negative coefficient, d(SP)/dt < 0.

Depending on the composition of feed water, boiler water may contain cations Ca²⁺ and Mg²⁺ and anions SO₄²⁻, SiO₃³⁻, PO₄³⁻, CO₃²⁻, and Cl⁻. As evaporation proceeds, the concentrations of all ions increase and approach the solubility limits of the substances involved.

Dissolved substances can crystallize from water. Those which have the lowest solubility product under particular conditions crystallize first. Some of them can crystallize on heating surfaces and form scale; these are called scale formers. Others crystallize in the bulk of the solution and form sludge and are thus called sludge formers. Rough elements (protrusions and recesses) on solid heating surfaces can serve as centres of scale formation, and disperse and colloidal particles and gas bubbles suspended in water can serve as centres of sludge formation.

The conditions required to avoid scale formation from Ca and Mg cations and SO_4^{2-} anions present in water can be written in the following form:

$$a_{\text{Ca}^{2+}} \cdot a_{\text{SO}_{4}^{3-}} < \text{SP} \atop a_{\text{Mg}^{2+}} \cdot a_{\text{SO}_{4}^{3-}} < \text{SP}$$
 (14.3)

In order to determine the allowable concentrations of Ca (or Mg) and scale-forming anions in water, it is essential to know the solubilities in water of all the substances which may form under particular conditions and their dependence on temperature. Such data for the principal scale formers are given in Fig. 14.1. As may be seen, they have negative temperature coefficients of solubility, so their solubility in water at high temperatures is only a few mg/kg, i.e. threefive orders of magnitude lower than that of easily soluble salts. When the solubility of a particular impurity, say, of CaSO₄, at a given temperature is known, one can determine the active concentrations of respective ions Ca2+ and SO2- and then calculate tho solubility product from formula (14.2).

Products of corrosion of structural materials. Feed water can bring into the steam-water path of boilers the corrosion products of a number of structural materials, such as iron, copper, zinc, cobalt, aluminium, etc. Their compounds can form deposits on boiler surfaces which are determined by their solubility in the aqueous medium under the process conditions. The solubility of most corrosion pro-

ducts does not exceed a few hundredths of a gram por kilogram* (Fig. 14.3).

The highest quantity of corrosion products enters the steam-water path in the form of oxides of iron which is the main structural material of boiler plants of any pressure. Iron combines with oxygen into a number of oxides, two of them, hematite Fc.O. and magnetite Fe₃O₄, being the most important in steam-generating plants. Of highest interest are the properties, solubility in particular, of magnetite as it is the main oxide of iron that can form at temperatures below 550-570°C, i.e. at the operating temperatures in high-pressure and supercritical-pressure plants.

The solubility of magnetite in hightemperature water depends substantially on the pH index of the medium. The solubility curves of magnetite in boiling water depending on temperature and at various values of pH (in oxidizing and reducing media) are illustrated in Fig. 14.2. As may be seen, in a wide range of high temperatures $250-350^{\circ}$ C (4-18 MPa) and pH = 6-9, the solubility of magnetite in reducing media does not exceed 40-50 μg/kg, in oxidizing media it does not exceed 20-30 μg/kg. The actual concentrations of this substance in water are much higher, which means that water contains colloidal and disperse particles of iron oxide, as well as dissolved iron oxides. Suspended (colloidal and disperse) iron-oxide particles, irrespective of their size, can form deposits on heating surfaces. The heaviest deposits (a few hundred grams per m2 in a year of operation) can form on the elements of the steamwater path which are heated most intensively. 70-90% of these deposits are iron oxides.

Deposits of corrosion products of iron usually have two layers possessing different physico-chemical properties. The *interval layer* is dense and bonded

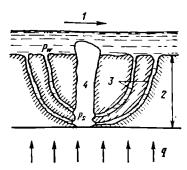


Fig. 14.5. Diagram of circulation in a deposited iron oxide layer

1—water flow; 2—thickness of deposited layer; 3—capillaries; 4—steam outlet from a 'steam pipe' at pressure p_{8} ; p_{40} is the pressure of water

firmly to the metal; it forms through corrosion on the metal surface. The external layer is loose and porous and only weakly bonded to the surface. The internal layer is not dangerous to operation of the metal and is even desirable, since the dense firm oxide film protects the metal from further corrosion. On the contrary, the loose porous external layer (which forms mainly from colloidal and disperseparticles) has a low conductivity and thus impairs heat removal from the metal surface.

In addition to the above, bubble boiling can lead to the 'wick effect' in porous iron-oxide deposits: water is sucked in through numerous capillary pores in a layer to the heating surface where it evaporates. Steam is then ejected back through a wide channel, or 'steam pipe' (Fig. 14.5). With such local circulation and evaporation of water, impurities (including corrosion-active impurities, such as alkalies, chlorides, etc.) are concentrated at the metal surface and can enhance corrosion.

In porous deposits, an appreciable portion of heat is removed due to the evaporating effect. This determines a high 'effective heat conductivity' of deposits, which includes heat conductivity as a physical costant and heat transfer from the wall to the working fluid. For this reason, the temperature of the metal in 'wick

At typical plI values of boiler water at thermal and nuclear power stations (see Ch. 15).

boiling' increases by not more than 10-20 deg C oven under substantially thick deposited layers (a few hundredths or even tenths of a millimetre). Heat transfer may be sharply impaired if capillary pores are clogged due to deposition of other impurities present in water.

Since porous deposits have a rough surface and an appreciable thickness, they can diminish the free crosssectional area of tubes and increase their hydraulic resistance, which lowers

the working pressure.

We can divide the formation of the external layer of iron-oxide deposits into three stages: the transport of suspended particles from the flow core into the layer at the wall; the motion of particles in that layer; and their attachment to the surface. At the first stage, the largest contribution is from hydrodynamic forces. At the second and third stages, depending on particular conditions of the process, forces of electrochemical nature may be active in addition to hydrodynamic and intermolecular forces. Electrochemical forces cause motion of the charged particles of corresion products in the electromagnetic field which appears in the heated layer of the heat-transfer agent just at the wall under the action of a thermo-e.m.f. The thermo-e.m.f. appears in the circuit consisting of the heating surface (first-order conductor) and agueous heat agent (second-order conductor) at a temperature difference existing between its portions. Due to these processes, the corrosion products suspended in water are deposited on the heating surfaces; at the same time they may be partially washed off from these surfaces by the working fluid.

The rate of formation of deposits is an important characteristic which determines the possibility of long uninterrupted operation of steam-generating plants. As applied to iron-oxide deposits, the rate of their formation depends on a large number of process parameters: mass velocity, heating

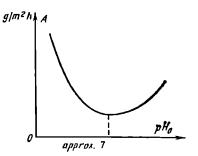


Fig. 14.6. Effect of pHo on the rate of deposition of iron oxides

load, boiling conditions, pH of aqueous heat agent, particle size and dispersity, etc. In view of the complexity of the process of iron-oxide deposition and its dependence on many factors, there is still no physical model which embraces the effects of all the parameters indicated. The effect of the pH index of heat agent on the rate of deposition of the ironoxide external layer at subcritical pressures and O2 concentration at a level of 0.4 mg/kg is shown in Fig. 14.6. As may be seen, the lowest rate of deposition is at the pH value corresponding to the isoelectric point of corrosion products.

As noted earlier, as suspended corrosion products are deposited on heating surfaces, they are partially washed off by the working fluid. At the beginning of the process, when the surface is free from deposits, the direct process prevails and the rate of deposition is at a maximum (Fig. 14.7). As a layer is accumulated on the surface, it is washed off more intensively. In the course of time, and

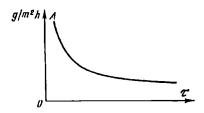


Fig. 14.7. Variation of the rate of formation of iron-oxide deposits

depending on the particular conditions, a balance is established between deposition and washing-off, and therofore, a particular rate of deposition. Products washed off from some portions of the surface can be deposited on other portions. This property of deposits has an adverse effect on the operation of circuits in nuclear power stations since it is one of the causes of radioactivity transfer along a circuit.

Easily soluble substances. Figure 14.4 shows the solubility characteristics of easily soluble impurities. The temperature coefficients of solubility in the region of interest for steam boilers and steam generators (above 200°C) are positive for some of them (NaOH) and negative for others (Na₂SO₄, Na₂PO₄).

When hydrodynamic and heat-transfer processes occur properly and ensure reliable temperature conditions on the heating surfaces, the concentration of each of the substances in the water of boiler drums is only a small fraction of the allowable value. For instance, at a water temperature of 343° C (p == 15.5 MPa), the solubility of Na₂SO₄ is roughly 10 g/kg water, i.e. five times the concentration allowed in boiler water to produce clean steam (2 g/kg). This example shows that the precipitation of a solid phase from a solution is only possible at a very high degree of vaporization of the solution in a boundary layer of boiling liquid, which is never achieved under normal hydrodynamic conditions in drum-type boilers. Therefore, under such conditions, easily soluble salts in water present no danger of deposit formation on the heating surfaces.

Solubility of impurities in working fluid at supercritical pressures and formation of deposits. The temperature of a process has an essential effect on the solubility of substances in H₂O at supercritical pressures, especially in the region of high heat capacities. Data obtained at a pressure of 25 MPa are illustrated in Fig. 14.8.

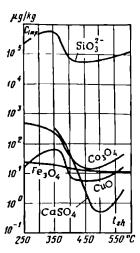


Fig. 14.8. Isobars of the solubility of selected substances in II₂O at supercritical pressure (25 MPa)

By comparing the solubility curves with some thermophysical parameters of aqueous heat agent, say, density p or dielectric permittivity e at supercritical pressures (see Fig. 9.6), it can be seen that they have essentially the same pattern. Like density or dielectric permittivity, the solubility of most impurities decreases inversely with the temperature in the whole temperature range of interest.

These regularities suggest that most of the impurities present in the working fluid of steam boilers at supercritical pressures can precipitate only in a rather narrow range of variations of thermophysical parameters, namely, in the region of high heat capacities (see Sec. 9.5). An exception to this rule is the solubility of the corrosion products of iron, mainly of magnetite Fe₃O₄, which varies only slightly with temperature (Fig. 14.8). Besides, it is only slightly dependent on the density of the heat agent, and therefore, on the state of the working fluid. For this reason, iron corrosion products tend to 'spread' along the wholo steam-water path of the boiler and turbine. On the other hand, the thickness of iron-oxide deposits de-

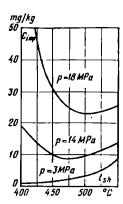


Fig. 14.9. Solubility of NaCl in superheated steam at subcritical pressures

pends substantially on the intensity of heating, so that the surfaces heated more intensively (such as the lower radiation sections of once-through boilers) are fouled with iron-oxide deposits much more substantially than those operating at lower heating rates.

Solubility of impurities in superheated steam of subcritical pressures and the formation of deposits. The solubility of substances in superheated steam of subcritical pressures is determined by the properties of the solvent (superheated steam) and the properties of the solid impurity with which steam is in contact, with both being dependent on the process parameters, i.e. pressure and temperature.

The strength of the bonds between ions, molecules or atoms of a solid impurity depends substantially on temperature. With an increase in temperature, these bonds are weakened and the solid phase can pass over into steam. Pressure variations in the range of operating pressures in steam boilers have little effect on the behaviour of the solid phase.

Temperature and pressure which determine the density of superheated steam, have a strong effect on its ability to dissolve solid impurities. At a constant pressure as the temperature of superheat increases, the steam density decreases, resulting in a dec-

rease in the diclectric permittivity of H₂O and a lower polarity of water molecules. As a result, the dissolving power of superheated steam first decreases with increasing temperature due to a decrease in its density. With a further increase in temperature at a constant pressure, however, although the steam density continues to decrease, the crystalline bonds in the solid are weakened and the solubility of the substance respectively increases. Figure 14.9 shows isobaric (constantpressure) curves of the solubility of NaCl depending on temperature. As may be seen, the effect of both factors (pressure and temperature) at the minimum of solubility is roughly the same. In the left-hand branches of the curves, the effect of temperature variations in density is predominant, while the pattern of the right-hand branches is determined by bond forces in the crystalline lattice.

The pressure of superheated steam also has a strong effect on its dissolving power. At higher pressures, steam has a higher density, and therefore, a higher dissolving power, but the effect of pressure lessens with an increase in the temperature of superheating. The isobaric solubility curves of other compounds, say, Na₂SO₄ or CaSO₄, have essentially the same pattern, but the quantitative relationships are different.

The practical significance of the solubility of substances in superheated steam consists in the following: if the concentration of an impurity in steam is lower than its isobaric solubility, it will be dissolved and carried off by steam and will usually form deposits in the turbine. If, however, its concentration is higher than the isobaric solubility, the excess will be deposited in the superheater path and the remainder in the turbine.

14.3. Passage of Impurities from Water to Saturated Steam

There are two known ways for impurities to pass from water to steam: they can be carried off as droplets of boiling water or by dissolution in steam.

The concentration of impurities in saturated steam can be characterized by the total carry-off coefficient k_{c0}^{o} , %:

$$k_{co}^{t} = \omega + k_{d} = \frac{C_{s}}{C_{10}}$$
 (14.4)

where ω is the water content of steam, %, which characterizes the concentration of impurities which can pass into saturated steam with water droplets, k_d is the distribution coefficient, %, which characterizes the concentration of impurities due to the dissolving power of steam.

The relative role of the components in the carry-off coefficient depends on a number of factors, primarily on pressure. For instance, at low pressures, the dissolving power of steam for most non-volatile impurities present in water is negligible $(k_d \ll \omega)$, and therefore, $k_c^i \approx \omega$. With an increase in pressure, the dissolving power of steam increases, resulting in a higher contribution by the distribution coefficient, so that it may turn out at a sufficiently high pressure that $k_d \gg \omega$ and $k_c^i \approx k_d$.

The mechanism and laws of water carry-off by steam. The mechanism of the formation of water droplets in the steam space of a boiler drum may be different depending on the scheme of steam supply. If steamwater jets are ejected below the water level into the drum, steam bubbles rise to the disengagement surface and form a two-phase dynamic layer (Fig. 14.10a and b). A steam bubble is acted upon by two forces: the force of internal pressure which tends to rupture the water film that envelops the bubble and the force of surface tension of that film which resists the rupturing force. In pure water, water flows down from the upper spherical surface of the film which consequently becomes thinner (Fig. 14.10c). A hole forms in the top of the sphere, which is widened by the forces of surface tension. The film is retracted into

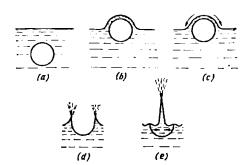


Fig. 14.10. Formation of condensed moisture in the steam space of a drum with evaporating tubes entering the water space (a) rising steam bubble; (b) bubble at the disongagement surface; (c) before break-through of water film; (d) and (e) break-through of water film and formation of moisture droplets

the bulk of the water, and the steam bubble is freed and enters the steam space. As an annular wave forms in the process, water droplets break off from its edge and are ejected into the steam space (Fig. 14.10d). Water tends to fill in the newly formed crater and water currents collide with one another in the centre and rise forming a vertical column from which water droplets can also break off (Fig. 14.10e). In steam bubbling through a layer of low-mineralized water, water films which envelop steam bubbles may have different thickness, and therefore, form water droplets of various

When steam enters the drum above the disongagement surface, water droplets can appear in the steam space due to disintegration of the moisture that is carried by the steam supplied from evaporating tubes (Fig. 14.11). The degree of water atomization depends on the kinetic energy of steam-water jets. At high heating loads, and therefore, a high velocity of exit of the steam-water jets into the drum, they possess a high kinetic energy and can atomize water into finer droplets, resulting in a more intensive water carry-off. A dynamic equilibrium is established in the steam space between the water droplets which enter the steam space and those which

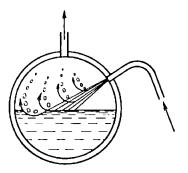


Fig. 14.11. Formation of condensed moisture in a drum with steam-water mixture entering the steam space

settle in the water. The concentration of water droplets is the highest just at the disengagement surface and decreases the farther it moves from that surface. The largest droplets can be ejected to a height of 600-700 mm.

If the rising velocity of steam is low, it can entrain only the finest water droplets. With an increase in the flow rate of steam, the larger droplets can be entrained. For this reason, the water content of produced steam turns out to be higher at higher heating loads (see Fig. 14.12).

The water content ω of steam is determined by the heating load D:

$$\omega = AD^n \tag{14.5}$$

where A and n depend on the design of the drum, pressure, and the concentration and ionic composition of impurities in the water. The exponent n changes substantially with the heating load. The dependence of the water content of steam on the heating load is approximated in logarithmic coor-

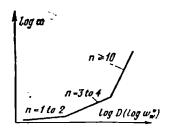


Fig. 14.12. Effect of the heating intensity on the water content of steam

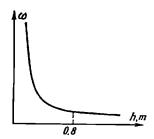


Fig. 14.13. Effect of the height of steam space on the water content of steam

dinates by broken straight lines expressed as power functions (14.5). There three such straight sections (Fig. 14.12). For loads characterized by a very low water content of steam $(\omega < 0.01\%), n = 1-2$; for loads at which $\omega = 0.01 - 0.1\%$, n = 3-4; and for higher loads with $\omega > 0.2\%$, $n \geqslant 10$. For drum-type boilers of thermal power stations, operation at the beginning of the second section is typical, for which n = 3-4. Steam velocity is proportional to its flow rate D. The average flow rate of steam related per m2 of the disengagement surface is called the rate of evaporation per m2 of water surface:

$$R_F = D/F \tag{14.6}$$

The average velocity of steam related to m³ of the steam space is called the rate of evaporation per m³ of steam space:

$$R_{\rm V} = D/V \tag{14.7}$$

The height of the steam space has a vital effect on the water content of produced steam. At $R_F = \text{constant}$, with a smaller height of the steam space, larger droplets can reach the region of high steam velocities at the inlet to steam-circulating tubes, and therefore, steam will contain more moisture. In the steam space of a larger height, the large droplets cannot reach the steam-circulating tubes as easily and the water content of produced steam will be lower (Fig. 14.13). Beginning from a certain height (roughly 0.8 m) which is

unattainable even for the most farreaching large droplets with the highest kinetic energy, any further increase in the height of steam space cannot significantly decrease the water content of steam. Under such conditions, steam carries off only fine droplets whose soaring volocity w_{so} is smaller than the rising velocity of steam w_w^* at $R_F = \text{constant}$. Fine droplets are transported by the steam flow irrespective of the height of steam space. The soaring velocity of water droplets is understood as the relative velocity of a droplet at which its mass is counterbalanced by the force of resistance in steam flow. For these conditions, it may be written:

$$\frac{\pi d_{dr}^3}{6} (\rho' - \rho'') g = \xi \frac{\pi d_{dr}^3 \rho'' w_{so}^2}{8}$$

whereby

$$w_{so} = 1.155 \sqrt{\frac{d_{dr}}{\xi} \left(\frac{\rho'}{\rho''} - 1\right) g}$$
 (14.8)

where d_{dr} is the diameter of droplet and ξ is the coefficient of resistance.

With an increase in pressure, the density of steam increases and offers a greater resistance to rising droplets. On the other hand, as the difference in the densities of water and steam decreases, the transporting ability of steam is enhanced. This is also due to the fact that an increase in pressure decreases surface tension, so that the size of the water droplets decreases and they can be more easily carried off by steam. Since, however, pressure has a stronger effect on the transporting ability of steam than on its resistance, an increase in pressure results in a higher water content of steam.

We have discussed the laws which govern the carry-off of droplets of pure or low-mineralized water. These laws hold true for a rather wide range of concentrations. All other conditions being identical, the water content of steam in this range is constant. Begin-

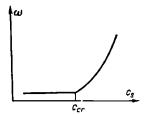


Fig. 14.14. Effect of salt content of water on water content of steam

ning from a certain concentration typical of a particular substance, the size of steam bubbles in water diminishes, and therefore, the velocity of their rise decreases and φ_{bub} increases. This results in the water swell in the drum and the ejection into the steam space of a large number of water droplets with a high concentration of the impurity, which critically impairs the steam quality (Fig. 14.14). The concentration of substances in water at which the water level suddenly swells up and increases the carry-off of moisture by steam is called the critical concentration.

Physico-chemical principles of the distribution of impurities between water and saturated steam in equilibrium. A two-phase single-component system may be, depending on the relative concentration of the phases, in the form of boiling water containing steam bubbles, or wet steam containing water droplets, or boiling water in contact with saturated steam. Irrespective of the structure of the twophase system, water and steam are essentially two solvents of the same chemical nature but different density and dielectric properties (see Sec. 9.5) which determine their ability to dissolve inorganic substances.

If the two-phase system is in thermodynamic equilibrium, the non-volatile impurities present in it are distributed between the phases according to the law of distribution of solutes in immiscible solvents. This equilibrium can be characterized quantitatively by the distribution coefficient k_d which is expressed in terms of the activity

of the solute in steam, a_s , and in water, a_w :

$$k_d = \frac{a_s}{a_{so}} \tag{14.9}$$

For dilute solutions (which are typical for the conditions of steam generation at thermal and atomic power stations), the activities can be replaced by the concentrations of the solute, i.e.

$$k_d = C_s^s/C_w^s$$
 (14.10)

The distribution coefficient depends on the form in which substances are present in an aqueous solution. Substances present in water in a molecular form have the highest capacity for passage from water to steam. Those which are present in an ionic form, will dissolve in steam much less readily. The law of distribution holds true for substances which are present in both solvents (steam and water) in one and the same form, either molecular or ionic. The distribution coefficients k_d^m and k_d^i corresponding to this condition are thermodynamically true and written as:

$$k_{d+tr}^{m} = \frac{C_{s}^{m}}{C_{w}^{m}}$$
 and $k_{d+tr}^{i} = \frac{C_{s}^{i}}{C_{w}^{i}}$ (14.11)

At a constant temperature, the true distribution coefficient of a substanco is constant and independent of the initial concentration of the substance in one of the phases. Determining the true distribution coefficients involves appreciable difficulties, since it is practically impossible to find separately the concentrations of substances in the molecular and ionic form. For this reason, the distribution coefficient is usually determined through the total concentrations of a substance without taking account of the forms of its existence in solvents, i.e. the apparent distribution coefficient is determined.

The correlation between the apparent molecular coefficient of distribution $k_{d,ap}^m$ and the true distribution

coefficient $k_{d,tr}^{m}$ is as follows:

$$k_{d,ap}^m = k_{d,tr}^m \beta$$
 (14.12)

Similarly, for the ionic form:

$$k_{d,ap}^{i} = k_{d,tr}^{i} (1 - \beta)$$
 (14.13)

where β is the fraction of the molecular form in the total concentration of a substance in the solution (β depends on pH and temperature) and (1 — β) is the fraction of the ionic form.

The total apparent distribution coefficient is the sum of the molecular and ionic apparent distribution coefficients.

According to the law of distribution between two immiscible solvents. the passage of substances from water to steam takes place under adiabatic conditions at a constant saturation temperature and particular pressure (density). On the other hand, as follows from Fig. 9.8, the density lines of water and steam at any temperature (pressure) have no real regions of gradual passage of substances from water to steam. This passage from water to saturated steam at equilibrium takes place not gradually, but suddenly, as follows from the distribution law. As pressure approaches the critical value, i.e. as ρ''/ρ' tends to unity, this "jump" declines and only at $p = p_{cr}$ can the substances dissolved in water pass over smoothly into saturated steam in equilibrium with water.

At low concentrations of substances in aqueous solution, the effect of process parameters on the coefficient of distribution between water and dry saturated steam in contact with it ($\omega = 0$) can be described by the equation proposed by Acad. M.A. Styrikovich:

$$k_d = \left(\frac{\rho''}{\rho'}\right)^n \tag{14.14}$$

which holds true for conditions when the solute is present in water and in steam in equilibrium with water in the same form (either molecular or ionic).

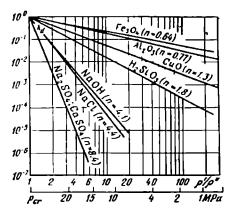


Fig. 14.15. Effect of the ρ'/ρ'' ratio of the aqueous heat transfer agent on the coefficient of distribution of substances

In equation (14.14), n is the coordination number which reflects the physico-chemical properties of the solute and expresses the degree of its hydration. It remains constant in a wide range of parameters. At a higher degree of hydration, the ionic form contributes more to the total concentration of the substance in solution. Such substances are rotained more firmly by polar water molecules and pass less easily into the steam phase.

The effect of the density ratio of steam and water on the distribution coefficients according to formula (14.14) can be represented in logarithmic coordinates in the form of a pencil diagram (Fig. 14.15). Pencils of all substances start from the origin of coordinates which corresponds to the critical pressure (for water, $p_{cr} = 22.85 \text{ MPa}$) and distribution coefficient $k_d = 1$.

All substances present in water can be divided into three groups according to their solubility in steam: (1) those for which n < 1; (2) those with n = 1-3; and (3) those with n > 4. Substances in the first group possess the highest solubility in steam. They include weak electrolytes — mostly the products of corrosion of structural materials: Fe₃O₄, Al₂O₃, etc. that may, however, have a low solubility

in water, for which reason their concentration in steam is usually not high. Of special interest is silicic acid H₂SiO₃, a weak electrolyte which belongs to the second group. It may be present in water in significant concentrations and possesses a high solubility in steam. Substances from the third group have the least solubility in steam; these mainly include salts and Na₂SO₄ and hydroxide NaOH, which are contained in water mostly in the ionic form and are especially sensitive to variations in the density of solvents (water and steam).

The apparent coefficient of distribution of weak electrolytes depends not only on the temperature (pressure) of the process, but also on the pli index of the aqueous solution which characterizes the degree of deviation from the predominantly molecular or predominantly ionic form of the substance. The form in which strong electrolytes exist, e.g. the salts of strong bases or acids in aqueous solutions, and their apparent coefficients of distribution vary with their concentrations.

If water contains both dissolved and undissolved, i.e. suspended, impurities which can pass into steam with water droplets, the total contamination of steam is determined from the equation:

$$C_s^t = (C_w^{tis} + C_w^{rus}) \omega + C_w^{tis} k_{d,qp}^t (14.15)$$

wherefrom the total coefficient of carry-over k_c^l , which characterizes the ratio of total concentrations of an impurity in water and steam is found from the formula:

$$k_{co}^{t} = \omega + \frac{C_{w}^{dis}k_{d.ap}^{t}}{C_{w}^{dis} + C_{w}^{sus}}.$$
 (14.16)

As follows from equation (14.16), the quality of steam depends on its water content ω and distribution coefficient $k_{a,ap}^f$. The water content of steam can be diminished by separation (see Sec. 15.4). Even the complete separation of moisture, however, can-

not make steam free from impurities. The distribution coefficient is a physico-chemical constant and, if water contains impurities, steam will have an equivalent quantity of these impurities in accordance with the condi-

tions of the process. In order to clean the steam from impurities which have passed into it from water due to the distribution law, it should be washed by cleaner water than that from which it has formed (see Sec. 15.4).



WATER CONDITIONS

15.1. Removal of Impurities from the Circuit

The reliable operation of steamturbine plants is ensured by maintaining the proper level of cleanliness of the working fluid, i.e. by removing impurities from the circuit in accordance with the rate of their passage to the water (see Sec. 14.1). Methods for removing impurities may vary depending on the type of boiler. Drum-type boilers operate under the principle of multiple forced or gravity circulation. The steam content of the fluid flow in the uptake tubes of circulation circuits is limited and does not normally exceed 10-25%. Boiler water does not evaporate deeply, and therefore, the impurities dissolved in it do not reach their extreme concentrations (to saturation) and thus cannot precipitate as a solid phase in the water bulk or on the tube walls. In order to retain the concentrations of impurities in water below the level at which they would precipitate in the solid state. part of the water is continuously removed from the boiler drum (blowdown water). Blowing-down as a means of removing impurities from power plants is especially efficient with those impurities which have a low coefficient of distribution between steam and water and for this reason are not properly carried off by steam (sodium salts and sodium hydroxide). Blowingdown is inefficient with impurities which have a high coefficient of distribution and are carried off in a large quantity by steam (silicic acid and metal oxides).

In once-through boilers, the evaporation process takes place with all the water being vaporized continuously. Blowing down once-through boilers is impossible, so impurities settle on the heating surfaces as deposits according to their solubility in water and steam. Easily soluble deposits which accumulate in particular zones of once-through boilers are partially washed off at the boiler starting-up and shutting-down. Slightly soluble deposits are removed periodically by chemical washing which is done after shutting down the boiler. The washing process is labour-consuming and requires much time and a large quantity of reagents. The continuous removal of impurities from the steamwater path of once-through boilers can be effected in a demineralizing plant arranged in the circuit downstream of the turbine condensers.

Feed water at the entry to the boiler contains a noticeable quantity of oxygen and carbon dioxide. Free oxy-

	Concentration of impurities, g/m ² , at heating loads, kW/m ²							
State of surface	up to 100	100-300	300-450	above 450				
Clean Requires cleaning	25-50 200-300	up to 25 150-200	up to 25 100-200	up to 20 100-150				

Table 15.1. Allowable Contamination of Internal Surfaces of Tubes

gen and carbon dioxide can cause intensive corrosion of the metal of boiler equipment. To prevent this, these elements are removed by thermal deaeration.

The water conditions at nuclear power stations are largerly determined by their operation specifics, i.e. by neutron irradiation of the heattransfer agent as it passes through the reactor core. Further, the corrosion products of structural materials are continuously accumulating in the circuit. If they are not removed in due time from the heat agent, they can form deposits on the surfaces of the circuit. These impurities are subjected to neutron irradiation in the reactor and become radioactive, thus creating a radiation hazard in the zone around the reactor equipment.

Since used water at nuclear power stations is radioactive, it cannot be blown down and discharged to escape channels, as is done at thermal power stations. As at thermal power stations, it is cleaned from impurities in ion-exchange resin water purifiers. This treatment prevents the formation of deposits on the working surfaces of the circuit.

The objectives of minimizing the corrosion of structural materials and the formation of deposits and of producing steam of a high purity (and, at nuclear power stations, minimizing the radioactivity of the heat-transfer agent) are achieved by properly organizing the physico-chemical processes in the steam-water path, i.e. by providing appropriate chemical water conditions*. It is virtually impo-

ssible to completely control corrosion. the activity of the working fluid and depositions on the working surfaces and produce absolutely clean steam (free from impurities). The optimal water conditions of a power unit should ensure the reliable uninterrupted operation of the equipment for a long time before chemical washing is required. The principal tasks are: to restrict the formation of internal deposits which might cause the temperature of heating surfaces to increase intolerably; to restrict the formation of deposits which might diminish the power of the unit in the flow path of the turbine; to suppress the corrosion of structural materials in the steamwater path and to minimize erosion wear.

Deposits on the surface of metal are removed by chemical cleaning of the equipment. This procedure is carried out in a new boiler before starting up and then periodically during operation. A washing circuit is assembled: for this purpose, which includes washing pumps, tanks for preparation of the reagents, connecting pipelines, and reservoirs for the collection and neutralization of wash water. The length of periods between chemical washing procedures depends on the operating conditions of the boiler plant, mainly on how carefully the specified water conditions are observed. The time for a chemical washing procedure is determined by the amount of deposits on heated tubes and by the intensity of their heating. A rough estimation of the allowable contamination levels of the internal surfaces of tubes is given in Table 15.1.

^{*} In further discussion, we will refer to this as water conditions.

15.2. Water Conditions of Once-through Boilers

An important factor in the organization of water conditions in oncothrough boilers is that water blowingdown is inapplicable. For this reason, all impurities which are brought in with feed water and those which pass into the working fluid due to corrosion of the boiler proper and in the feed water path behind the condensate cleaners are partly deposited on the heating surfaces and partly carried off into the turbine. It should be borne in mind that only very slight deposits are allowed in the turbine. The amount of deposits allowed in the boiler is many times that allowed in the turbine, i.e. a subcritical-pressure boiler should be regarded as a 'trap' for impurities which prevents their passage to the turbine. Under such conditions, the concentration of impurities in feed water can be somewhat higher than in superheated steam, accordance with the allowable level of deposits in the boiler. The allowable deposits depend on their distribution along the steam-water path, the heating load and conductivity in places of deposit formation, and on the reliability reserve of heating surfaces, i.e. how far the temperature of the metal may rise above the working temperature without causing creeping and scaling. Under identical conditions, the allowable amount of deposits depends heavily on the heating load, which should be taken into account when selecting the zone of deposition in the gas path. The zone of deposition is usually the final section of the evaporating path and it should not be arranged in the region of intensive heating. The length of the zone of deposition also depends on pressure. At higher pressures, the deposition zone is longer and begins at a lower steam content of the fluid flow.

In high-capacity boilers for supercritical pressures, furnace water walls operate at substantially higher heating rates and are therefore much more sensitive to deposits. Since deposits are unwanted in both the turbine and boiler, once-through boilers must be fed with water that is as free as possible of impurities. In practice, this means that all the condensate from the turbine must be cleaned in a demineralizing plant. This treatment almost totally prevents the deposition of salts and silicic acid in the boiler and turbine, while the problem of preventing the deposition of corrosion products, especially iron oxides, becomes more important.

A common method for increasing the corrosion resistance of equipment is to select the appropriate materials. The main structural material for making the heating surfaces of boilers is pearlitic steel. Pearlitic steels have many advantages (low cost, good workability, serviceability), but they possess a serious drawback: they are subject to intensive corrosion in the steam-water path of boilers.

Corrosion damage is reduced by the application of protective coatings to the internal surfaces of tubes in the low-temperature sections of power plants. Those include: the internal surfaces of atmospheric-type deaerating tanks, exhaust hoods of turbines, housings of condensers and low-pressure vacuum heaters, condensate tanks and their pipelines, and water-treatment equipment. In the high-temperature sections of power plants, corrosion is limited by appropriate organization of water conditions.

Hydrazine-ammonia water treatment. Thermal deaeration cannot remove oxygen and carbon dioxide completely from the turbine condensate. The concentration of residual oxygen may be as high as 10 μg/kg. Residual carbon dioxide is also present in the condensate. For this reason, thermal deaeration is supplemented with the chemical treatment of feed water.

Residual oxygen in feed water after thermal deaeration is bound by hydrazine N₂II₄. If water contains no other impurities, the reaction is as

follows:

$$N_2H_4 + O_2 \rightarrow N_2 + 2H_2O$$
 (15.1)

Feed water always contains impurities: iron and copper oxides. In their presence, oxygen is hound by hydrazine more quickly [38].

In order to ensure the complete removal of oxygen, hydrazine is fed at the intake to feed pumps in an amount exceeding the stoichiometric ratio from formula (15.1), i.e. an excess of hydrazine is formed in an amount of 0.02-0.03 mg/kg.

Carbon dioxide may be present in water in the form of either molecules CO₂ (dissolved gas) or II₂CO₃ (solution of carbonic acid):

$$CO_2 + H_2O \Rightarrow H_2CO_3$$
 (15.2)

Carbonic acid can be found by a measured quantity of ammonia added to feed water. Ammonia is introduced in an amount sufficient for the complete neutralization of CO_2 to form ammonium carbonates and to allow a slight excess of ammonium hydroxide to raise the pH index of water. Thus, hydrazine hydrate combines the residual oxygen in deaerated water, while ammonia maintains the pH index at the required level pH = 9.1 \pm \pm 0.1.

Hydrazine-ammonia water treatment, i.e. the chemical treatment of feed water by hydrazine hydrate and ammonia, is a conventional method and until recently was employed at almost all supercritical-pressure monobloc units.

The temperature of the working fluid at the outlet of the lower radiation sections of boiler plants is usually 380-390°C. The temperature of the outside surface of tubes in this section is roughly 100 degrees higher, i.e. attains 490-500°C. As established by experience, with the application of hydrazine-ammonia water treatment, the temperature of tubes in the lower radiation section of fuel oil-fired boilers increases by 10-12 deg C every month. For this reason, chemical washing should be carried out periodically in

4-6 months in order to retain the temperature of tube walls below the allowable level.

Neutral water conditions. All monobloc units of power stations are provided with demineralizing plants to produce clean feed water. Upon cleaning in a demineralizing plant, turbine condensate approaches the theoretically pure neutral water with the electric conductivity $0.04\text{-}0.06~\mu\text{S/cm}$ and pH index around 7. This water is practically pure and almost free of ionogenic impurities, so that all electrochemical processes in it are retarded.

Oxygen in neutral water may produce disferent essects on metals depending on its concentration. At low concentrations, it can enhance metal corrosion. At clevated concentrations of oxygen, a continuous film of magnetite Fe3O4 or hematite Fe2O3 forms on the metal surface. For this reason, it has been proposed to prevent further corrosion of metal by introducing oxygen into water in an amount (around 200 μg/kg) sufficient to form a passivating continuous film of iron oxides. Under such conditions, the corrosion rate of pearlitic steel is almost as low as that of austenitic steel.

The ability of oxygen (in elevated concentrations) to form firm protective oxide films has been utilized as the basis for the organization of oxygenneutral water conditions in once-through boilers. For this purpose, gaseous oxygen O₂ is added in a measured quantity to feed water. In some cases, hydrogen peroxide H₂O₂ is used instead of O₂. To form neutral water conditions, feed water should be very pure and contain no CO₂; its electric conductivity should not exceed 0.2 μS/cm.

Neutral water conditions are advantageous in the following respects: it can be dispensed with expensive correction treatment of feed water by hydrazine hydrate and ammonia; in that connection, filters of the demineralizing plant can operate for a longer time between regenerations; the rate of formation of iron-oxide deposits in

intensively heated surfaces of the lower radiation section is lower; and pearlitic steels can be employed in heating surfaces. In operation with neutral water conditions, care should be taken to keep the electric conductivity of feed water as low as possible. For this reason, the feed-water path should contain no elements made of copper or copper alloys. Neutral water conditions have been employed at a number of supercritical-pressure monobloc units for several years.

Chelate treatment of water. When iron-oxide deposits form on heating surfaces, the temperature of the metal is determined by the heating intensity and the properties of the deposits, mainly by their thermal conductivity, which is lower in porous deposits.

Temperature conditions on heating surfaces can be improved by two methods: by increasing the thermal conductivity of deposits or by causing the deposits to form primarily in the less heated elements (say, in the economizer), rather than in the lower radiation section.

The properties of iron-oxide deposits and the regularities of their formation in the steam-water path can be changed by employing the chelate treatment of water. The method of chelate water treatment has been proposed and developed by T. Kh. Margulova et al. and consists essentially in the following. Feed water is treated with ammonia and hydrazine hydrate added in the same quantities as in the conventional hydrazine-ammonia method. In addition, chelates are introduced into the deserated feed water in a quantity equivalent to the concentration of iron and copper in it. Chelates are substances capable of forming water-soluble compounds with various cations (Ca, Mg, Fe, Cu). Ethylene diamine tetraacetic acid (EDTA, dry product) is usually employed for the purpose.

An aqueous solution of the acid at a temperature of 80-90°C is prepared and mixed with an aqueous solution of ammonia: $NH_3 + H_2O = NH_4OH$.

The trisubstituted ammonium salt of ethylene diamine tetraacetic acid which forms on mixing can react with the products of iron corrosion at temperatures of 100-200°C, with iron hydroxide Fe (OII)₂ to form *iron chelates*. These are readily soluble in water and dissociate at a high temperature, with a dense layer of magnetite being deposited on the internal surfaces of tubes, protecting the metal against corrosion.

The most intensive dissociation of iron chelates occurs at temperatures of 250-300°C which are typical for the last stages of the high-pressure water heater and economizer, so that complete thermal dissociation (thermolysis) would be expected to take place exactly in these elements of the steam-water path. The working fluid, however, moves in this section of the path (high-pressure water heater and economizer) at a noticeable velocity. 3-5 m/s and 1-2 m/s, respectively. Morcover, in supercritical-pressure boilers, the temperature of the working fluid increases quite rapidly as it moves along the path (at a rate of roughly 100°C/min). For these two reasons, the thermolysis of chelates cannot come to its end in the economizer and partially continues in the next element of the path, the lower radiation section. Therefore, iron-oxide deposits in supercritical-pressure boilers are distributed in the following manner: around 80% settle in the economizer and 20% in the lower radiation section. On addition of the chelates, the deposits in the lower radiation section become denser and have a higher thermal conductivity, which lowers the rate at which the temperature of the wall rises in time and makes it possible to prolong the period between washings up to 18 months.

It is advisable to automatically add the chelates according to the heating load of a monobloc unit. Ammonium salt of ethylene diamine tetraacetic acid and ammonia are added to the feed water after the deacrator, and hydrazine after the demineralizing plant. The concentrations of additives in feed water in chelate water treatment are as follows: E1)TA = $= 80 \mu g/kg$, NH₃ = $700-800 \mu g/kg$, N₂11₄ = $20 \mu g/kg$; pH = 9.1.

Gascous products formed on the thermal dissociation of the chelates pass together with steam through the turbine and are removed from the cycle as exhaust from the condenser. Chelate water treatment offers certain advantages typical for 'neutral' water conditions (increased time between washings), but also possesses the drawbacks of the hydrazine-ammonia treatment (an increased load on the demineralizing plant, since a large quantity of ammonia should be removed from the circuit, and a high use of reagents for filter regeneration).

15.3. Non-scaling Water Conditions of Drum-type Boilers

Drum-type boilers are often fed with softened water, i.e. water that contains easily soluble substances, mainly sodium salts.

Owing to inleakage of the cooling water in condensers, calcium and magnesium salts can also get into the feed water. These salts have a very low solubility (milligrams and tens of milligrams per kg water), which further decreases with increasing temperature. During steam generation at a high pressure, the concentrations of these salts readily rise to a value at which they can form scale. Not all calcium and magnesium compounds form scale; some of them form sludge. Some kinds of sludge [such as Mg₃(PO₄)₂] can stick to heating surfaces, which is also undesirable. Calcium and magnesium can form non-sticking sludge [for instance, $3Ca_3$ (PO₄)₂·Ca(OH)₂ or $3MgO \cdot 2SiO_2 \cdot 2H_2O$] which remains in the boiler water in a suspended state and can mostly be removed by continuous blowing-down. A small amount of heavier sludge accumulates in the lower headers and can be removed by periodic blowing-down.

In order to correct water conditions and transfer the hardness salts into non-sticking sludge, it is essential to introduce correction additives to water, for instance, phosphates (such as sodium phosphate Na₃PO₄) to fixed calcium. Water conditions based on the addition of phosphates are called phosphate water conditions. For easier formation of non-sticking movable sludge, phosphates should be added to an alkaline medium. For this reason, sodium phosphate is introduced not into the feed water which has a low alkalinity, but into the boiler drum where the alkalinity of the water is sufficiently high due to multiple evaporation. Thus, phosphate-alkaline water conditions are formed. The reaction of sludge formation can be written in the following form:

The reaction products are removed by blowing-down. For the complete removal of calcium salts, a certain excess of phosphates is maintained in the boiler water. The excess of PO³-for boilers without stepped evaporation is 5-15 mg/kg; for those with stepped evaporation, the excess of PO³-is 2-6 mg/kg for the clean section and not more than 30-50 mg/kg for the salt section.

Due to the hydrolysis of phosphate ions that takes place in phosphate treatment, hydroxy ions form which further increase the water alkalinity:

$$\frac{PO_4^{3-} + H_2O \approx HPO_4^{3-} + OH^-}{HPO_4^{3-} + H_2O \approx H_2PO_4^{3-} + OH^-}$$
 (15.4)

Upon phosphate-alkaline treatment, the hydrate alkalinity of water may sometimes turn out to be substantial (pH > 11) and cause metal corrosion.

In boilers supplied with turbine condensate with an addition of chemically purified water, alkalinity is held at a moderate level by adding not pure Na₂PO₄, but a mixture with an acid phosphate, such as Na₂HPO₄.

At power stations fed with turbine condensate and low-mineralized make-up water (chemically desalted water or distillate from evaporators), water alkalinity is controlled only by the hydrolysis of phosphates [see formulae (15.4)], i.e. by forming water conditions of purely phosphate alkalinity. Phosphates are continuously introduced into the boiler drum by metering pumps.

In recent times, the quality of feed water at power stations has improved appreciably due to lower inleakage into condensers, thus resuting in a higher quality turbine condensate. This makes it possible to employ water conditions with a lower degree of phosphate treatment or even to dispense with phosphate treatment and change to no-phosphate non-scaling water conditions. No-phosphate water conditions make the hoiler operation less expensive (since corrective additives are not used), decrease the salt content in boiler water, improve the quality of steam, and lower the cost of boiler equipment.

15.4. Methods for Generating Clean Steam

The purity of steam should satisfy extremely high requirements. For instance, the total concentration of impurities in superheated steam of supercritical pressure must not exceed 40-50 µg/kg. Methods for producing clean steam vary depending on the type of plant.

In once-through boilers, the working fluid (water), is continuously vaporized, so that its impurities are partly deposited on heating surfaces and partly carried off with steam. With an increase in pressure, the concentration of impurities in steam increases and the quality of steam approaches that of feed water (Fig. 15.1). Blowingdown is inapplicable in once-through boilers. The sole way to make clean steam is to improve the quality of feed

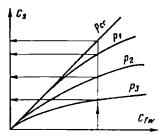


Fig. 15.1. Effect of pressure on the steam quality in a once-through boiler $(p_{cr} > p_1 > p_2 > p_3)$

water. Thus, the quality of steam produced in once-through boilers is controlled by specifying the quality of feed water.

In drum-type boilers, the purity of saturated steam, and therefore, of superheated steam is determined by the quality of the water from which it is produced. With a lower concentration of impurities in boiling water (under identical conditions), cleaner steam can be obtained. Blowing-down of drum-type boilers can improve the quality of circulating water, though excessive blowing-down can diminish the efficiency of the steam-turbine plant owing to heat loss with blow-down water.

Separation of moisture droplets from steam. To produce clean steam, it is essential, first of all, to dry it as completely as possible, i.e. to separate water droplets from the steam flow. Steam-separating systems should satisfy the following main requirements: low water content in the produced steam, high unit steam-generating load, and low hydraulic resistance.

The separation of moisture from steam is based on the density difference between water and steam. A water droplet in the steam space of a boiler drum is acted upon by two opposite forces: the lifting force and the force due to gravity. The relationship between these forces and the time of their action determine whether a droplet will be carried off by steam or settle on the water surface. It is clear that a higher separating effect can be

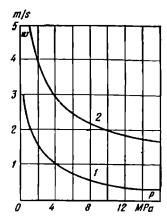


Fig. 15.2. Effect of pressure on the steam velocity in the holes of a submerged distribution plate

t- minimal velocity; 2-recommended velocity

achieved at a lower rising velocity of steam in the drum.

Among the simplest and most efficient devices for steam separation are perforated plates made of steel, with the perforations 5-12 mm in diameter. A perforated plate is mounted 100-150 mm below the average water level in the drum; it is called a submerged plate (as in steam generators of nuclear power stations). Another plate is arranged in the steam space at the top of the drum (perforated baffle). Both plates serve to equalize the distribution of steam over the drum cross section. For uniform bubbling of steam through the submerged plate, continuous steam layer (steam cushion) should be formed beneath the plate. The condition of steam cushion stability (see Sec. 13.1) is determined by steam velocity in the perforated plate holes. This velocity depends on pressure, i.e. it is lower at higher pressures (Fig. 15.2).

In high-capacity boilers, each evaporating tube delivers into the drum up to 1 000 kg/h of steam-water mixture on the average, some tubes deliver up to 1 500 kg/h. These powerful flows ejected into the drum possess a high kinetic energy which should be dampened in order to minimize steam

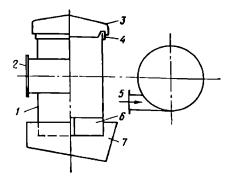


Fig. 15.3. Internal cyclone steam separator thousing: 2—inter pipe connection; 3—cover; collar; 5—steam-water mixture in; 6—cross-piece; thousand the transfer of the connection of the cyclone cyclone in the cyclone cyclone

moistening. This is performed in steam separating devices.

The principal steam-separating device in high-capacity steam boilers is a cyclone separator which is arranged inside the boiler drum. An internal cyclone separator (Fig. 15.3) is essentially a cylindrical vertical housing 300-400 mm in diameter into which steam-water mixture is introduced tangentially at a speed of 6-8 m/s. Upon entering the cyclone, the steam flow kinetic energy produces a centrifugal effect, and the flow is whirled at the cyclone surface. Water is pressed against the walls and flows downward, while steam flows from the cyclone uniformly at a velocity around 1 m/s from beneath the cyclone cover into the steam space of the drum (Fig. 15.4). A cross-piece in the cyclone bottom stratifies the water flow which moves smoothly into the water space of the drum. The number of cyclones in a drum is determined by the capacity of the cyclone which in turn depends on its dimensions and pressure: the capacity of a cyclone 300 mm in diameter at a pressure of 4 MPa, 10 MPa and 15.5 MPa is respectively 4 t/h, 6 t/h and 10 t/h.

Internal cyclone separation is rather effective, but increases the hydraulic resistance of the circuit, which should be considered in circulation calculations.

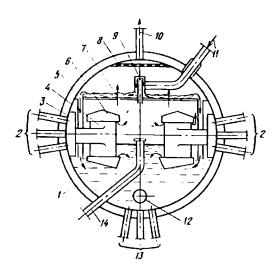


Fig. 15.4. Typical drum internals 1—drum; 8—evaporating tubes; 3—box; 4—cyclone; 5—drainage box; 6—cover; 7—steam-washing perforated plate; 8—perforated steam baffle; 9—feed water distribution box; 10—steam-circulating tubes; 11—feed water in: 12—hole in a partition; 13—downtake tubes; 14—emergency water drainage

External cyclones have also found wide application for steam separation in drum-type boilers; they are mounted outside the boiler drum. An exter-

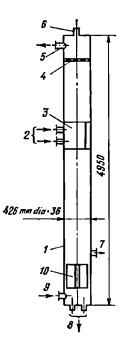


Fig. 15.5. Typical external cyclone separator

1—header; \$\mathcal{s}\$-steam-water mixture in; \$\mathcal{s}\$-whirler; \$\mathcal{s}\$-steam-outlet pipe; \$\mathcal{s}\$-air vent; \$7-blow-down; \$\mathcal{s}\$-downtake tubes; \$9-water from drum; \$10-cross-piece

nal cyclone separator (Fig. 15.5) is made in the form of a vertical cylinder 350-450 mm in diameter. The steamwater mixture is introduced tangentially and the separation process is essentially the same as in an internal cyclone separator. The height of an external cyclone is determined by the sum of the required heights of the steam space (1.5-2.5 m) and water space (2-2.5 m) to ensure efficient separation and stabilize the operation of the downtake tubes in the circulation circuit which are connected to the external cyclone.

In once-through boilers, the steamwater path is provided with a starting-up unit mounted between the furnace water walls and the subsequent heating surfaces. One of its principal elements is a built-in separator (for more detail see Sec. 23.4). It has a vertical cylindrical housing with a single stage of steam-separating vanes inside; the steam-water mixture is introduced at the top (Fig. 15.6). Water droplets are thrown to the walls by centrifugal forces, water flows through an annular drainage chamber and is removed through a side drainage tube. Dried steam passes from the separator through the bottom chamber. This design of separator has been unified. has standard dimensions and is em-

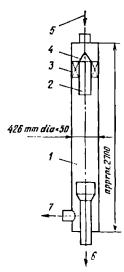


Fig. 15.6. Unified built-in (start-up) steam separator

1—header; 2—whirler insert; 3—whirler; 4—distributing cone; 3—steam-water mixture in; 6—steam out; 7—drainage of separated water

ployed in supercritical-pressure monobloc units for 500, 800 and 1 200 MW.

If wet steam moves in a tube at a moderate velocity, its moisture precipitates and flows along the walls as an annular film. This process is called film separation. For effective separation of the flow into steam and water, the flow velocity should not be excessively high, otherwise water droplets will break off from the film and he carried off by steam. The highest allowable velocity depends on the pressure and steam content of the flow (Fig. 15.7).

At present, turbines of nuclear power stations operate mostly on saturated steam. Therefore, to attain higher efficiency, the moisture content of steam must be as low as possible, which is achieved by careful primary separation with a slight superheat before the low-pressure cylinder of the turbine. The process diagram of separation is determined by the type of reactor.

In channel-type graphite-water reactors, saturated steam is separated in horizontal drums outside the reactor.

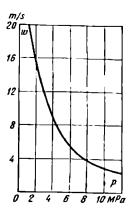


Fig. 15.7. Effect of pressure on the steam velocity at which water droplets can be broken off from the liquid film surface

A powerful reactor may have a number of separating drums connected in parallel. For instance, the reactor unit type RBMK-1000 (1 000 MW) has four separating drums. The drums are made of carbon steel Grade 22K (0.22 C boiler), with the entire internal surface being plated by stainless stool. The drum is 31 m long and has a diameter of 2.3 m. The connection diagram for the separating drums of an RBMK-1000 reactor is illustrated in Fig. 24.13. The number and dimensions of separating drums are not related to the dimensions of the reactor and are chosen so as to attain the highest degree of steam drying. Soparating drums of the design described can produce steam with a water content of not more than 0.1%.

In water-moderated water-cooled tank-type reactors, steam separation is carried out in the reactor heusing (see Fig. 24.12). Precipitating separation in a small volume is insufficiently effective. The steam-water mixture formed in the fuel assemblies of the reactor core is fed into a collecting box and then through parallel vertical tubes into axial cyclones. Upon separation, water is returned into the water space, while steam rises through the central portion of the cyclones into a steam drier and then is

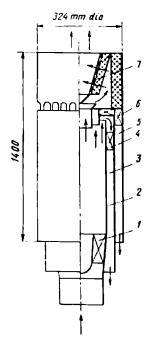


Fig. 15.8. Primary centrifugal separator of axial type

1—whirler; 2—cyclone; 3—channel for primary removal of separated moisture; 4—flow-straightening vanes; 3—channel for the secondary removal of separated moisture; 6—separating vanes in the path of secondary moisture; 7—predrier

delivered through steam pipelines to the turbine.

A feasible version of a primary centrifugal separator for tank-type reactors is shown in Fig. 15.8. Under the action of centrifugal forces developed in a whirler, the steam-water mixture is separated into the peripheral water layer which moves along the internal cylindrical wall of the cyclone, and the steam flow which moves through the cyclone core. The main mass of water is removed from the cyclone via the primary drainage channel which is provided with straightening vanes to stop rotation of the water flow. The remaining water is removed through a secondary drainage channel in which vanes are mounted to separate the steam which enters this channel together with water. The separated steam is directed together with the main steam flow into a preliminary drier and then into the main drier.

A steam drier usually has a set of vertically-arranged corrugated stainless-steel plates (Fig. 15.9). Each bend of plates has a welded-on corner plate which forms a vertical pocket for trapping and removing moisture.

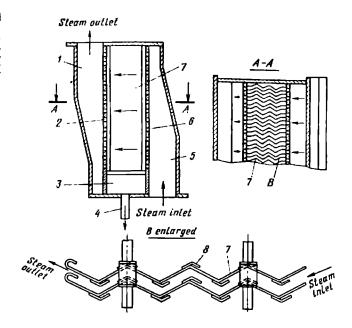
Steam washing. Re-writing equa-(14.4) in the form: $=C_{w}(\omega+k_{d})$, it may be seen that steam quality can be improved by either improving the quality of feed water (which involves extra expenditures on water treatment), or decreasing the water content of steam (achieved by separation of moisture from steam flow), or by decreasing the distribution coefficient. The distribution coefficient is the constant of equilibrium between boiling water and saturated steam and depends on pressure and on the physico-chemical properties of substances dissolved in water. For a particular dissolved substance. given pressure and given concentration of the impurity, the distribution coefficient is constant. In turn, $k_d =$ $= C_{\kappa}^{dis}/C_{w}^{dis}$, and therefore, cleaner steam can be produced at $k_d = constant$ by decreasing the concentration of impurities in water, which again involves extra expenditures on water treat-

Noting, however, that the purity of delivered steam is determined not by the water from which it is generated, but mainly by the water with which it contacts on entry into the steam space of the drum, it is possible, with feed water of a given quality, to substantially reduce the concentration of impurities in steam by washing, i.e. by passing it at the last stage of the process through a layer of pure water, say, condensate or pure feed water. This is accomplished in bubble-cap steam washers.

Steam washing by bubbling consists essentially in the following (Fig. 15.10). Suppose that dry steam of a salt concentration C_{s1} is generated from boiler water with a high concentration of salts C_{bw} . In this process,

Fig. 15.9. Saturated steam drier

1 and 5-constant-velocity channels; 2 and 6-perforated plates; 3-horizontal cluste; 4-drainage pipes; 7-corrugated plates; 3-moisture-trapping pocket



an equilibrium will be established according to the dissolving power of the steam with respect to a given substance, which depends only on the process parameters. The equilibrium state is characterized by the distribution coefficient $k_1 = C_{s1}/C_{bw}$. As steam passes further through a layer of feed water with a low salt concentration C_{fw} , a new equilibrium state is established and the steam will have another salt concentration corresponding to the distribution coefficient $k_2 = C_{s2}/C_{fw}$.

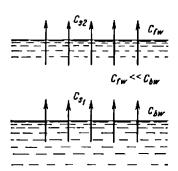


Fig. 15.10. Principal diagram of steam washing

As the new equilibrium is established, the dissolved impurity in steam will partly pass over into water, since $C_{*2}/C_{fw} > k_2$, and therefore, its concentration in the steam will decrease and that in the wash water, increase. With a good, sufficiently long contact between the two phases, $C_{*2}/C_{fw} > k_2$.

Since both processes (steam generation and washing) occur at the same pressure, $k_1 = k_2$. Furthermore, since C_{bw} is much greater than C_{Jw} . it turns out that C,2 is much lower than C_{st}. A strong washing effect can be achioved by passing steam in fine jets through a water bed, for instance, through a perforated plate. Upon washing, saturated steam is subjected to secondary separation to reduce its moisture content roughly to its prowashing level. The simplest steam-washing device is a perforated plate onto which wash water is poured (Fig. 15.11). For efficient operation, water should not be allowed to pass through the perforations, especially at low loads. This is attained by maintaining a definite steam velocity (see Sec. 13.2).



Fig. 15.11. Steam washer

Steam washing is carried out in the steam of a boiler drum (see Fig. 15.4). The edges of the perforated plate are bent up to retain the required layer of water. The dimensions of the washing device are such that the internal diameter of the drum must not be less than 1 600-1800 mm. Feed water is used for steam washing. The flow rate of wash water is determined by the steam-generating capacity. In modern steam boilers, all feed water is fed into a distributing box and the excess of water flows through a slit in the box directly into the water space of the boiler drum, i.e. it does not participate in steam washing.

Stepped evaporation. The balance of salts for the simplest version of the water conditions of a drum-type boiler with blow-down (Fig. 15.12) is as follows:

$$DC_s + D_{bd}C_{bd} = (D + D_{bd}) C_{fw}$$
 (15.5)

Dividing both parts of equation (15.5) by D and denoting $D_{bd}/D = p$, we obtain:

$$C_s + pC_{bd} = (1 + p) C_{fw}$$
 (15.6)

wherefrom

$$C_{bd} = \frac{(1+p) C_{fw} - C_s}{p}$$
 (15.7)

Neglecting the salt content of steam $(C_* \approx 0)$ and assuming, for instan-

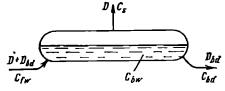


Fig. 15.12. Simplified diagram of organization of water conditions in a hoiler drum C_{fw} . C_{bw} . C_{bd} . C_s —concentrations of substances respectively in feed water, botter water, blowdown water, and steam

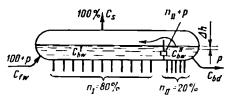


Fig. 15.13. Diagram of two-stage evaporation

Designations the same as in Fig. 15.12; additionally: n_1 and n_{11} —steam-generating capacities of the first and second evaporation stages, percent of the steam-generating capacity of the plant

ce, that the blow-down ratio p = 1%, we have:

$$C_{bd} = C_{bw} = \frac{(1 \pm 0.01) C_{fw}}{0.01} = 101 C_{fw}$$
 (15.8)

As follows from this equation, steam is generated under the given conditions from boiler water with the salt content exceeding that of feed water by 101 times. The quality of steam can be improved by more intensive blowing-down, but this is economically inefficient. For continuous blowing-down, the blow-down ratio is taken within 0.5-1.0% if the water loss is replenished by distillate from evaporators or by demineralized water, and 0.5-3% of chemically purified water is added.

A more efficient method is that of stepped evaporation proposed by E. I. Romm. The boiler drum is divided by a partition into two compartments (Fig. 15.13). Each of them is connected to a group of circulation circuits not interconnected by the water path. The water spaces in the two compartments communicate through a small hole provided in the partition. Feed water is fed into the first (larger) compartment and blowingdown is effected through the second (smaller) compartment. Boiler water can flow through the partition hole from the first into the second compartment, but the water level in the latter always remains below that in the former. Steam is removed from the drum only through the first compartment. Thus, the device shown in Fig. 15.13 has two evaporation stages. Assuming, for instance, the relative steam-generating capacity of the first stage $n_1 = 80\%$ and that of the second stage, $n_{11} = 20\%$, the concentrations of impurities in boiler water will be respectively as follows:

in the 1st evaporation stage:

$$C_{bw}^{1} = \frac{\frac{[n_{1} + (n_{11} + p)] C_{fw}}{n_{11} + p}}{= \frac{(80 + 20 + 1) C_{fw}}{20 + 1}} = 4.8C_{fw}$$
(15.9)

and in the 2nd stage:

$$C_{bw}^{11} = \frac{(n_{11} + p) C_{bw}^{1}}{p}$$

$$= \frac{(2^{(1-|-1)}) 4.8C_{fw}}{1} = 101C_{fw}$$
(15.10)

As may be seen, in two-stage evaporation, C_{bw}^{1} is much lower than C_{bw}^{11} , and for this reason the first compartment, where the salt content of water is not high, is called the pure compartment and the second, where water has a high salt content, is called the salt compartment. The ratio C11/C1 is called the concentration ratio. In the above example, 80% of the total quantity of steam are generated from water having a low salt concentration, and therefore, the main mass of steam produced has a higher quality than in the scheme with one evaporation stage, and only 20% of the total quantity of steam are generated from water of the same quality as in the single-stage method. Therefore, the quality of steam in two-stage evaporation is substantially better than in single-stage evaporation.

The overflow of water from the pure to the salt compartment is essentially internal blowing-down of the former. In contrast to external blowing-down, internal blowing-down involves no losses of heat, nor of the working fluid, so that the blow-down ratio can be chosen solely from considerations on how to obtain the best quality

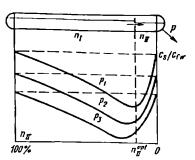


Fig. 15.14. Determination of the optimal capacity of salt compartment

Blow-down ratio: $p_1 < p_2 < p_1$

of steam. In turn, the blow-down ratio determines the steam-generating capacity of the salt compartment, which can be obtained from the appropriate calculation. Assuming various values of steam-generating capacity of the salt compartment, n_{11} , one can determine the quality of saturated steam produced at a selected value of p of external blowing-down. It is clear that at $n_{11} = 0$ and $n_{11} = 100\%$ there will be no stepped evaporation and the quality of steam in these extreme cases will be the same and correspond to the highest contamination. In the whole region of stepped evaporation $0 < n_{11} < 100\%$, the quality of steam will be higher (it will have a lower content of impurities) than in a simple single-stage scheme (Fig. 15.14). One can calculate the optimal steamgenerating capacity n_{11} at which steam will have the lowest concentration of impurities. Each particular value of p in external blowing-down has a corresponding optimal value n_{11} .

With internal stepped evaporation in boiler drums, the difference in the water levels in the compartments is not large in view of the limited height of the water and steam space of the drum. This may cause a reverse overflow of water. If this difference is increased by raising the water level in the pure compartment, the height of the steam space will diminish, and therefore, the carry-off of water droplets will increase. A decrease in the water

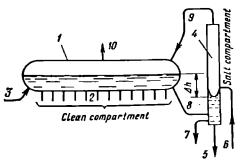


Fig. 15.15. Two-stage evaporation circuit with an external cyclone

1—drum; 2—evaporating tubes of pure compartment; 3—feed water in; 4—salt compartment (external cyclone); 5—downtake tube; 6—evaporating tubes of salt compartment; 7—blow-down; 8—feed line of salt compartment; 9—cross-over steam tube; 10—steam-circulating tube

level in the salt compartment may disturb the circulation.

With the use of external steamseparating cyclones, the difference in

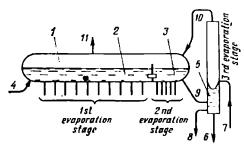


Fig. 15.16. Three-stage evaporation circuit with an external third stage

1—drum; 2. J. and 5—1st, 2nd and 3rd evaporation stages; 4—feed water in; 6—third-stage downtake tube; 7—third-stage evaporating tubes; 8—blowdown; 9 and 10—cross-over tubes for water and steam; 11—steam-circulating tube

the water levels in the compartments will be high enough to avoid a reverse overflow of water. For this reason, schemes with external cyclones are preferred, especially at a low capacity of the salt compartment.

The effectiveness of stepped evaporation increases, though not in proportion, with the number of stages. Two-and three-stage schemes have found wider application. The second evaporation stage can be arranged either inside the boiler drum as in Fig. 15.13, or outside, i.e. in external cyclones (Fig. 15.15). In a three-stage scheme, the first and second stages are usually arranged in the boiler drum and the third in an external cyclone (Fig. 15.16).

In external cyclones, the steam and water space may be of any height. This ensures thorough drying of the steam (due to a high steam space) and reliable operation of the circulation circuits (due to a high water space), and prevents water carry-over from the salt to the pure compartment.

Stepped evaporation makes it possible to improve the purity of steam at a given quality of feed water and given blow-down ratio. Furthermore, steam that is adequately pure can be obtained from water of a lower quality, which makes the water-treatment system simpler and less expensive. Additionally, stepped evaporation can increase the efficiency of a steam-turbine plant, since the blow-down ratio can be reduced without a significant loss in steam quality.



PROCESSES ON THE FIRESIDE OF HEATING SURFACES

16.1. Mechanism of Scaling

Various mineral impurities which pass into boiler furnaces together with the organic mass of solid fuels

are transformed in the high-temperature zone: part of them are melted and combined into larger particles which fall onto the furnace bottom as slag, while the main mass of fine particles of ash are carried off from the furnace by combustion products. The behaviour of ash particles in the furnace and flue ducts of boiler depends on their composition and physical properties (melting point, viscosity, thermal conductivity, etc.).

The composition of ash includes a small quantity of low-fusible compounds, mainly chlorides and sulphates of alkali metals [NaCl. Na₂SO₄, CaCl₂, MgCl₂, Al₂ (SO₄)₃] with a melting point of about 700-850°C. They are vaporized in the high-temperature zone of the flame core and then condense on the surface of the tubes, since the temperature of walls of clean tubes is always less than 700°C.

Medium-fusible components of ash, with a melting point of 900-1 100°C (FeS, Na₂SiO₃, K₂SO₄, etc.), can form the primary sticky layer on waterwall and platen tubes if the high-temperature zone is too close to the tubes owing to improper organization of combustion (the flame touches the tubes).

High-melting compounds of ash are, as a rule, pure oxides (SiO₂, Al₂O₃, CaO, MgO, Fe₂O₃, etc.). Their melting point (1 600-2 800°C) exceeds the highest temperature in the flame core, so that they pass through the combustion zone without being changed, i.e. they remain solid. Since their particles are small, these components are mainly carried off by the gas flow and constitute what is called fly ash.

In the zone where the temperature of gases is still high (above 700-800°C), low-fusible compounds are first condensed from the gas flow on the surface of clean tubes and form the primary sticky layer on them. At the same time, solid (high-melting) ash particles adhere to this layer. They solidify and form the primary dense layer of slag which is firmly bonded to the tube surface. The temperature on the outside surface of the layer increases and further condensation from gases is thus stopped. The rough outside surface of that layer retains fine solid particles of high-melting ash

which fall on it; these form an external loose layer of deposits. Thus, deposits on the tube surface in this region of gas temperatures most often have two layers: a dense internal layer and a loose, or friable external layer.

Ash particles can quickly form growing deposits on furnace tubes in the zones where high-temperature gases contact water walls. First, ash and slag particles in a semi-liquid or softened state are thrown onto the surface of the tubes; upon cooling, they adhere firmly to the surface. The process is called *slagging*. Slag aggregations may be quite large in size and have a mass of up to a few tons. The presence of relatively easily fusible particles in the combustion zone is determined by the formation of eutectics between metal oxides MeO (such as CaO, MgO, FeO or Fe₂O₃) and silica SiO, or alumina minerals based on Al₂O₃. The ash of most solid fuels contains 5 to 40% of metal oxides MeO. A higher concentration of MeO lowers the softening temperature of ash and creates the risk of slagging. However, if ash contains more than 80% of $Al_2O_3 + SiO_2$, its melting point rises substantially; it becomes high-melting.

With an unfavourable mineral composition of fuel (if the content of calcium oxide CaO is more than 40%), sintering (sulphation) may start in a deposited friable layer formed on the heating surfaces if SO2 is present in the flue gases. The process results in the growth of dense, firmly bonded slag deposits on the tube surface (Fig. 16.1). Slagging may occur on the tubes of water walls, platens, and sections of the convective superheater in the region of gas temperature up to 600-700°C. Caked deposits may overlap intertubular spacings of a width of up to 400 mm. Horizontal and slightly inclined tubes are slagged more intensively than are vertical tubes.

In zones of relatively low gas flow temperatures (less than 600-700°C) which are typical of heating surfaces in the convective shaft, friable depo-

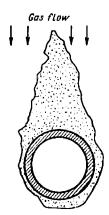


Fig. 16.1. Caked deposits on the surface of a tube

sits are more likely to form. There is no dense underlayer on the surface of tubes, since the condensation of alkali metal vapours has already been finished before that region.

Loose deposits chiefly form on the back side of tubes relative to the direction of gas flow, i.e. in the turbulent zone that forms behind the tube (Fig. 16.2). On the front side of the tubes, loose deposits can only form at low velocities of the flow (less than 5-6 m/s) or when the flow carries very fine fly ash.

In analysing the formation of friable deposits, ash particles are divided into three groups according to their size [17]. The first group includes the finest fractions (called inertialess particles) which are so small that they move along the flow lines of gases. Therefore, the probability of their settling onto the tube surface is low. This group includes particles of a size of up to $10~\mu m$.

Coarse fractions, more than 30 µm in size, are placed in the second group. These particles possess a sufficiently high kinetic energy to destroy loose deposits on contact.

The third group includes ash fractions of a size between 10 µm and 30 µm. When gases flow around tubes, these particles settle chiefly on tube surfaces and form a layer of deposits.

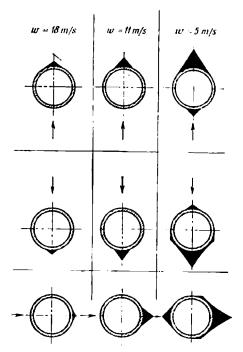


Fig. 16.2. Formation of loose deposits on tubes at various directions and velocities of gas motion

In the final result, the thickness of loose deposits is determined by the dynamic equilibrium between the processes of continuous settling of medium fractions and those of destruction of the settled layer by coarser particles.

Loose deposits on tube surface impair heat transfer, which is estimated by the *fouling coefficient*:

$$\varepsilon = \delta_{al}/\lambda_{al} \qquad (16.1)$$

where δ_{sl} and λ_{sl} are the average thickness and conductivity of the slag layer over the tubo perimeter.

The fouling coefficient ε , (m² K H)/W characterizes the thermal resistance of slag layers. The fouling of tubes with fly ash deposits depends only slightly on the concentration of ash in the gas flow. A difference in the rate of deposition may be observed only a few hours after starting-up until a dynamic equilibrium is established.

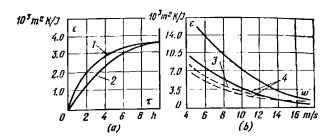


Fig. 16.3. Fouling coefficient for tubes depending on the concentration and particle size of ash in gas flow (tubes: d=38 mm, $s_1/d=s_2/d=2$)

(a) depending on the time of operation; (b) depending on flow velocity; I- ash concentration 21 g/m³; 2- 7 g/m³; 3- fine ash (mesh residue $R_{10}=24.5\%$); 4-coarse ash (mesh residue $R_{10}=52.5\%$)

(Fig. 16.3a). On the other hand, the size composition of ash has a strong effect on the formation of deposits. Finer fractions of ash cause more intensive fouling of tubes with a thicker layer of deposits being formed (Fig. 16.3b).

The degree of tube fouling depends substantially on the gas flow velocity. The amount of medium-size particles that settle on tubes increases roughly in proportion to flow velocity. The destructive effect of coarse particles, on the other hand, increases in proportion to the third power of velocity, the total result being that the thickness of the deposits on the tubes decreases with an increase in gas velocity. As demonstrated by experiments (Fig. 16.2), the intensity of tube fouling in a cross flow does not depend on the direction of flow. Under comconditions, vertical tube parable coils are less subject to fouling.

The degree of tube fouling heavily depends on the type of tube bundles (staggered or in-line) and the longitudinal pitch s_8 of tubes in staggered bundles. Under comparable conditions (the same gas velocity and tube diameter), the fouling coefficient for staggered bundles turns out to be 1.7-3.5 times that for in-line bundles (Fig. 16.4).

Tube fouling increases significantly if the gas velocity is less than 3-4 m/s. For this reason, the operation of heating surfaces of boilers at such low velocities is unadvisable. Since

the load of a boiler can drop in operation by as much as 50% of the rated capacity, the nominal gas velocity at the rated load should be not less than 5-6 m/s.

In boilers fired on high-sulphur fuel oil, both sticky deposits and dense glassy deposits can form on the heating surfaces in the zones where the gas temperature is below 600°C. Sticky deposits on the heating surfaces of convective superheaters and economizers primarily contain vanadium compounds (mainly V_2O_5) and sulphates. Dense deposits consist mainly of iron sulphates and calcium and sodium oxides. Deposits that form during fuel oil burning have a tendency to grow up quickly, which may greatly impair heat transfer, increase resistance of the gas path, and shorten the campaign of the steam boiler plant.

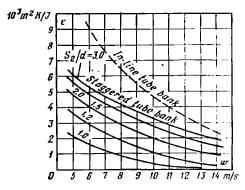


Fig. 16.4. Comparison of fouling coefficients for various tube bundles

Since fuel-oil ash deposits contain a noticeable quantity of vanadium and sulphur, they are essentially acid. The addition of certain substances with alkaline properties to fuel oil makes ash deposits more friable. The same effect is obtained by a particular combustion process, for instance, at an excess air ratio close to unity [17].

Some methods of cleaning heating surfaces from deposits are based on the dynamic effect of jets of steam. water or air. The effectiveness of such jets is determined by their range, i.e. by the distance at which a jet still has a sufficient dynamic head to destruct deposits. Water jets have the greatest range and the highest thermal effect on dense deposits. Devices based on this effect are used for cleaning furnace water walls. Water blowing should, however, be calculated carefully so as to avoid sharp cooling of the tube metal upon removal of the deposits.

Multi-jet retractable blowers have found wide application for cleaning radiation heating surfaces and convective superheaters. They operate on saturated or superheated steam supplied at a pressure of around 4 MPa.

Platens and in-line tube banks are cleaned by vibration cleaning: high-frequency vibrations transferred to heating tubes disturb the bond between the tube metal and deposits. Vibration cleaning is carried out by means of vibrators attached to water-cooled rods.

The most efficient method for cleaning convective heating surfaces in the downtake shaft of boiler furnaces is shot-blasting, which utilizes the kinetic energy of flying 3-5 mm cast-iron shots. Shot is carried upwards in the air flow and distributed over the whole cross section of the shaft. The use of shot is determined from the optimal intensity of shot-blasting, usually 150-200 kg/m² of a convective shaft cross section. The shot-blasting procedure usually takes up 20-60 s.

Shot-blasting is effective if it is regularly carried out soon after boiler start-up when the heating surfaces are still relatively clean. Recently, the heat-wave cleaning method has found use. It is based on the application of acoustic low-frequency waves generated in a special explosive-combustion impulse chamber.

Regenerative air heaters arranged outside the boiler are cleaned by blowing their packing with superheated steam (at a temperature 170-200 deg C above the saturation point). Less frequently, water washing (water removes sticky deposits, but increases corrosion) or shock-wave cleaning and thermal cleaning are omployed. Thermal cleaning is based on periodically raising the temperature of the regenerator packing up to 250-300°C by interrupting the air supply to the apparatus. In this way, sticky doposits are dried up and sulphuric acid is vaporized.

16.2. Abrasion Wear of Convective Heating Surfaces

The steam boiler designer should pay serious attention to measures preventing abrasion wear of the tubes of heating surfaces by particles of ash and unburned fuel. An improper selection of gas velocities in the flue duct of the convective shaft can lead to intensive wear of tube metal in certain places, thinning of the walls, and even break-through of the tubes.

The mechanism of abrasion wear is essentially as follows. Coarse particles of ash, which may be quite hard and have sharp edges, impinge on the walls of a tube and continuously cut off microscopic layers of metal. Thus, in critical places the wall is gradually thinned. Particles of unburned fuel (in particular, of harder grades, such as anthracite or semianthracite) can also cause abrasion wear of the surfaces.

Thus, wear of the boiler tubes is determined first of all by the abrasiveness of ash particles, which, in turn, depends on the content of SiO_2 in ash and increases noticeably when $SiO_2 > 60\%$. Wear intensity also de-

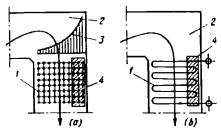
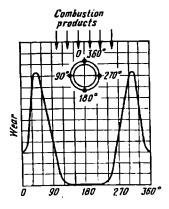


Fig. 16.5. Distribution of coarse fractions of fly ash behind a turning chamber and zones of dangerous abrasion wear of metal of the heating surfaces

(a) coils parallel to the boiler front; (b) coils perpendicular to the boiler front; I--coils; 2--turning chamber; 3--distribution of coarse ash fractions; 4--zone of abrasion wear of tukes

pends on the total ash content of fuel, A^{rc} .

The intensity of abrasion wear of heating surfaces may he uneven over the cross section of a gas duct in which the heating surface is located or around the tube perimeter. It increases significantly in places where the gas flow turns through 90°, say, on entry into the convective shaft (Fig. 16.5). Coarser ash particles are thrown against the rear wall of the shaft, which leads to greater wear of the tubes arranged on that wall. In the cross flow of staggered tube bundles, the greatest abrasion wear occurs on the front portions of tubes at an angle of incidence of gas flow of 30-50° (Fig. 16.6). In in-line tube bundles, the intensity



F.g. 16.6. Abrasion wear of a tube in cross flow

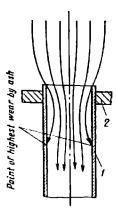


Fig. 16.7. Abrasion wear of a tube in longitudinal flow in an air heater

1-tube: 2-upper tube plate

of wear is substantially lower, since some tubes are in an aerodynamic shadow behind those in front of them. With the longitudinal flow of gases in tubes (as in air heaters), abrasion wear occurs in the inlet portion of a tube at a length of 150-200 mm due to impingement of coarse particle on the wall just after constriction of the gas jet (Fig. 16.7). Farther in the tube, the flow is stabilized and coarse particles move parallel to the tube walls.

Convective tube banks should have no longitudinal gas channels in which gas velocity might increase substantially. Wear on the outer bends of a tube is especially dangerous, since the tube wall in such places is thinner.

The intensity of wear is determined by the following factors:

(1) the kinetic energy of ash or fuel particles, which is proportional to the square of gas velocity w_e^2 ;

(2) the number of particles (concentration μ_{ash}) passing along the surface per unit time, which depends on the ash content of fuel and is an increasing function of w_k ;

(3) non-uniformity of ash concentrations in the gas flow, k_{μ} , and of gas velocities in the cross section, k_{w} ;

(4) the density of tubes in a bundle, i.e. the relative tube pitch $s_i d$.

In the final result, the intensity of wear, mm/year, depends on the third power of gas velocity:

$$I_{we} = amk_{\mu}\mu_{ash} (k_w w_g)^3 \left(\frac{s_1 - d}{s_1}\right)^{1.8} \tau$$
(16.2)

where a is the coefficient of ash abrasiveness, mm s³/(g h), m is the index of wear resistance of tubes, which depends on the steel composition, and τ is the time of operation of a surface, h.

For the normal operation of a tube for at least 10 years ($\tau = 60\,000$ -100 000 h), the allowable wear of its wall is $I_{we} = 0.2 \, \text{mm/year}$.

For staggered carbon-steel tube bundles with $s_1/d=2.5$, the allowable gas velocities determined by the normal wear by the ash of various fuels are as follows:

Fuel	•	•	•	•		u,	Ŗ٠	m/s
Ekibastuz coal								
Moscow district coal								9.0
Chelyabinsk coal .								10.0
Kizel coal								
Anthracite grade ASI	ì							11.5
Donetsk coal grade T			•	•	•			12.0

To select the proper gas velocity in the gas duct of a steam boiler, the most economically favourable gas velocity w_{ec} (see Sec. 20.6) should be compared with the allowable velocity w_{we} determined by the conditions of wear. If it turns out that $w_{we} > w_{ec}$, the latter can be used in calculations. Otherwise, the velocity of gases should be limited by the conditions of wear, which necessitates an increase in the dimensions of heating surfaces and flue ducts.

In any case, the risk of abrasion wear should be minimized by taking proper measures to prevent wear of tubes. These include placing steel cuffs on the tube in the places where increased wear is most probable, such as tube bends and mounting cut-in inserts in the inlet portions of air heater tubes, etc. These devices are detachable and can be easily replaced when the boiler is shut down for repairs and inspection of heating surfaces.

16.3. Corrosion of Heating Surfaces

High-temperature corrosion. The term 'high-temperature corrosion' implies corrosion damage of the metal of tubes which are in contact with hightemperature combustion products $(\vartheta_g > 700$ °C). It includes two kinds of corrosion which occur in various zones of the boiler and are of different chemical nature. One of them is the corrosion of water walls in the boiler furnace in the zone of the flame core, which is fostered by the contact of tube metal with sulphurous gases. Another kind is the corrosion of superheater tubes and their fastening elements in the presence of vanadium oxides in the gas flow.

Corrosion on the fireside of water walls occurs in boilers fired by pulverized coals with a low yield of volatiles (anthracites, semianthracites, lean coals) and high-sulphur fuel oil. This kind of corrosion develops intensively on tubes at the level of furnace burners or slightly above it in the zones directly swept by flame. Under unfavourable conditions, the rate of corrosion wear on the front portions of tubes may be as high as 3-4 mm/year, i.e. water wall tubes in this zone get out of order in less than one year (with the wall thickness 5-6 mm).

As established experimentally, the main corrosion-active component in furnace gases is hydrogen sulphido H₂S. Even with a slight volume concentration of H₂S at a surface (0.04-0.07%), the rate of metal corrosion increases roughly tenfold. At temperatures of 1 400-1 600°C, hydrogen sulphide burns in the presence of oxygen almost instantly. Therefore, it can be present in the zone near the water wall only in a reducing medium where oxygen is locally deficient.

The primary product of the reaction of H₂S with tube metal is ferrous sulphide FeS which is then changed to ferrous sulphate and flakes off from the tube surface, thus exposing the tube metal to further corrosion.

To avoid corrosion damage to water walls, fuel and air should be distributed evenly between the burners so that the excess air ratio in each burner is greater than unity. It is advisable to keep the flame from striking the water walls; this is achieved by arranging the extreme (corner) burners farther from the side walls or by turning them towards the furnace centre.

Corrosion of the tubes of convective superheaters has been detected on the combustion of fuel oils when the temperature of the tube walls exceeded 610-620°C. This kind of corrosion, called vanadium corrosion, is caused by the vapours of vanadium pentoxide V₂O₅ formed in furnace gases. If fuel oil contains sodium oxide, the combustion products will contain sodium vanadates (5V₂O₅·Na₂O·V₂O₄) which have a low melting point $(t_{mp} \approx 600$ °C). If the temperature of the superheater tubes is above 610°C, sodium vanadates can form a liquid film which is corrosive for various steels (carbon, low-alloyed and austenitic steels). Uncooled fastening and spacing elements of tubes whose temperature is close to that of the gases are also subject to heavy vanadium corrosion.

Corrosion is enhanced in the presence of sulphur oxides in the gas flow. Sodium pyrosulphates Na₁S₂O₇ are most dangerous for the metal; in combination with V₂O₅, they make the medium highly corrosive even at temperatures near 600°C. The highest rate of corrosion is observed at 700-750°C and is typical for tube-fastening elements.

The rate of vanadium corrosion can be diminished by giving various alkali additives to fuel oil, such as an aqueous solution of MgCl₂ in a concentration of 0.6-0.8 kg per t fuel oil. The most efficient method, however, is to keep the temporature of superheater tubes below 600°C.

Low-temperature corrosion. This kind of corrosion occurs on the heating surfaces of air heaters which ope-

rate at relatively low temperatures of the gases and working fluid (air).

The decisive factor in intensive lowtemperature corrosion is the presence of sulphuric acid vapours in the flue gas flow. As the sulphur of the fuel burns in the flame core, it forms sulphurous anhydride SO₂. With a certain surplus of air, SO2 is further oxidized to SO_n by atomic oxygen O" that forms in the high-temperature zone of the flame due to chain reactions of combustion and thermal dissociation. Sulphuric anhydride can dissociate only at rather high temperatures. The resulting reaction of formation and dissociation of SO₃ in the flame zone can be written as follows:

$$SO_2 + O' \xrightarrow{h_1} SO_2 \xrightarrow{h_2} SO_2 + \frac{1}{2}O_2$$
 (16.3)

where k_1 and k_2 are the reaction rate constants of the direct and reverse reaction, with k_1 being greater than k_2 . As a result, SO_3 appears in a noticeable concentration at the boundary of the flame core, but dissociates on completion of combustion. Its concentration then gradually decreases.

As the temperature of gases in the gas path gradually decreases, the process of SO_3 dissociation is retarded and is practically stopped at $\vartheta_g=1\ 200-1\ 250^{\circ}$ C. Thus, with quicker cooling of the gases in the gas path, the residual concentration of SO_3 will be higher.

As the gases pass further through convective heating surfaces, the concentration of SO_3 may even rise. Depositions on heating surfaces, such as soot particles, may serve as catalysts for the afteroxidation of SO_2 to SO_3 . As a result, the concentration of SO_3 in the gases constitutes 1-5% of the initial content of SO_2 , or 0.002-0.010% of the total gas volume.

In the zones where the gas temperature drops down below 500°C, SO₃ begins to react with water vapours and forms sulphuric acid vapours which are carried by the gas flow. The process is completed at a temperature near 250°C.

Corrosion on heating surfaces may start if the temperature of the wall and the boundary layer at the wall turns out to be below the condensation point of water vapours or sulphuric acid vapours at their partial pressures in the gases. The temperature at which moisture is condensed on a solid surface is called the thermodynamic dew temperature (dew point) $t_{d,p}$. For pure water vapours at their partial pressure in combustion $p_{\rm H_{2O}} = 0.01 - 0.015$ MPa, $t_{d,p}$ = 45-54°C. If sulphuric acid vapours are present in the gas flow, the temperature of condensation (sulphuric acid dew point $t_{d,p}^s$) is substantially higher (up to 140-160°C).

For fuel oil combustion, the dew point can be roughly determined from the formula:

$$t_{d.p.}^{s} = t_{d.p.} + 250 V \overline{S^{r}O_{2}}$$
 (16.4)

where $S^r = S^w/Q_l^w$ is the resolved sulphur content of fuel, % kg/MJ, and $O_2 = \frac{21(\alpha-1)}{\alpha}$ is the concentration of surplus oxygen in the gas flow, %.

With a higher sulphur content of fuel and higher excess air ratio α, more SO₃ forms in gases, resulting in a higher dew point.

Figure 16.8 presents the diagram of phase equilibrium between liquid and vapour in the two-component system $H_2O-H_2SO_4$ at various partial pressures of water vapours. The lower curves at p = constant describe the boiling

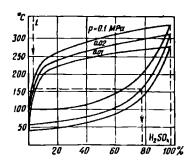


Fig. 16.8. Phase equilibrium of H₂O-H₂SO₄ system at various pressures

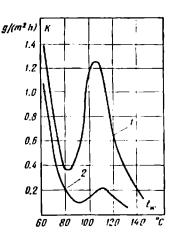


Fig. 16.9. Effect of temperature on the corrosion rate in packing sheets of regenerative air heater

t—high excess air ratio in furnace $(\alpha_f \le 1.1)$; 2—lowest allowable excess air ratio $(\alpha_f = 1.02$ -1.03)

temperature of an aqueous solution of sulphuric acid as a function of acid concentration, and the upper ones describe condensation temperature (dew point) of vapours. As may be seen, even a slight concentration of $\rm H_2SO_4$ vapours in flue gases (the dotted line on the left) sharply increases the condensation temperature, and the liquid film formed on the wall has a high concentration of sulphuric acid.

Figure 16.9 shows two typical curves of corrosion rate at different temperatures on low-temperature heating surfaces in contact with the flue gases obtained on combustion of high-sulphur fuel oil. As may be seen, the rate of corrosion varies non-monotonically with the wall temperature (Fig. 16.9, curve I). As t_w decreases from the dew point (around 145°C), corrosion first increases sharply to a maximum at $t_w = 105-110^{\circ}$ C, then falls off steeply, and finally, at temperatures of the wall below 85-90°C, there is a second rise in the rate of metal corrosion.

Corrosion of metal occurs in the presence of a condensed liquid film containing H₂SO₄ on the metal surface. It can continue further if new portions

of H₂SO₄ are supplied from the gas flow. Thus, the rate of corrosion is proportional to the condensation rate of H₂SO₄ vapours.

The intensification of corrosion at temperatures below 85-90°C is determined by the effect of the solution of sulphurous acid H₂SO₃ on the metal, which forms at low temperatures by the combination of H₂O and SO₂ on the liquid film surface.

The rate of corrosion in the temperature range of 80-120°C decreases substantially at a decrease in the excess air ratio (curve 2 in Fig. 16.9), which is associated with a less intensive formation of SO₃ and H₂SO₄ vapours in the flue gases in that temperature range. The same effect on the corrosion rate is obtained with a decrease in the sulphur content Sw in fuel. In this case, the maximum of corrosion rate as a function of wall temperature t_w retains its original position. The increased rate of corrosion at tw below 80°C does not significantly depend on air conditions, since it is determined by the effect of sulphurous, rather than sulphuric, acid on the metal.

The flow of gases produced by the combustion of solid fuel carries a large amount of fly ash, the basic properties of which are due to the presence of compounds of calcium and other alkali metals. As fly ash reacts with sulphuric acid vapours, the concentration of sulphuric acid in the gases decreases, resulting in a lower corrosion rate.

To avoid low-temperature corrosi-

on, it is essential that $t_w = t_{d,p}^{\ \prime}$, + + $(10\text{-}15)^{\circ}$ C, i.e. the wall temperature should be 10-15 deg C higher than the sulphuric dew point. This is economically feasible only in the combustion of low-sulphur fuel oils and sulphurous solid fuels, for which the sulphuric dew point $t_{d,p}^{\ \prime}$ does not exceed $100\text{-}110^{\circ}$ C. In other cases, the temperature of flue gases, which is determined by t_w , will be too high.

The lowest allowable temperature on the surfaces of a tubular air heater can be found from the formula:

$$t_w^{\min} = \frac{0.8\alpha_g \vartheta_g + \alpha_a t_a'}{0.95\alpha_g + \alpha_a} \quad (16.5)$$

where α_g and α_a are the coefficients of heat transfer at the gas and air sides of the heating surface, W/(m² K), ϑ_g and t_a' are the temperatures of the gases at the exit and the air at the infet, °C.

The coefficients 0.8 and 0.95 consider respectively the effect of tube fouling on the gas side and non-uniformity of the temperature field of gases over the cross section of the gas duct.

Under comparable conditions, the minimal wall temperature of regenerative air heaters is 10-15 deg C higher than that for tubular air heaters, since for the former α_g is roughly equal to α_a , while for the latter, $\alpha_a \approx 1.8 \alpha_g$.

Methods for increasing the wall temperature t_w of air heaters and reducing the rate of metal corrosion will be discussed in Ch. 19.



EVAPORATING HEATING SURFACES

17.1. Heat Absorption by Evaporating Surfaces and Their Layout

Evaporating (steam-generating) surfaces of various boilers may differ in design, but are always arranged, for the most part, in the furnace shalt and absorb radiant heat. The water walls of a boiler receive 35-40% of the total heat released in the furnace. This, in turn, has a strong effect on the distribution of heat between various heating surfaces (Table 17.1). For in-

Table 17.1. Heat and Temperature Distribution Between Boller Heating Surfaces

steam steam	team c:	Feed water temps-rature, c.	Heat distribution between heating surfaces, %					
Superheated a pressure, Miv	Superheated steam pressure, Mia Super cated steam temperature, *c.		evaporating	superheating	rconomizer			
4 10 14 14 25.5	440 540 570 570/570 565/570	145 215 230 230 260	62 49 39 32	19 30 36 46 58	19 21 25 22 42			

stance, at an average pressure of 4 MPa, the heat absorbed by radiant heating surfaces is insufficient to cover the total heat demand for steam generation (62%), so that part of the heat spent on water evaporation is transferred to the economizer. For this reason, the economizer of mediumpressure drum-type boilers is usually of the boiling type, i.e. water is not only heated in it to the saturation line, but also partially vaporized.

In high-pressure drum-type hoilers (14 MPa or more), the fraction of heat used for evaporation is considerably lower (see Table 17.1), so that the heat transferred in the furnace is sufficient to produce the required quantity of steam and the economizer can be of the non-boiling type. Once-through boilers are also provided with non-boiling economizers (see Sec. 11.2) from which water passes to the evaporating tubes through a distribution header. If the header were fed with a steamwater mixture rather than with water. this would cause an extremely uneven distribution of the mixture between the parallel tubes.

In medium-pressure drum-type boilers, additional evaporating surfaces are formed by a boiling economizer and, sometimes, by convective eva-

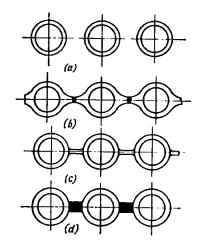


Fig. 17.1. Types of water walls

(a) with smooth (bare) tubes; (b) gas-tight with finned tubes; (c) gas-tight with welded-on rectangular fins; (d) gas-tight with metal building-up between tubes

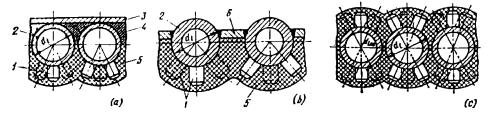


Fig. 17.2. Types of refractory-faced water walls

(a) smooth-tube vertical water wall; (b) membrane-type vertical water wall; (c) platen wall; 1—studs; 2—tube; 4—boller enclosure; 4—chromite paste; 5—carborundum; 6—cross-piece

porating surfaces or convective banks. In once-through boilers, convective evaporating surfaces form a transition zone in the convective gas duct, which resembles a tube-coil economizer; it is arranged between the superheater and economizer. In the transition zone, evaporation ends and the steam is slightly superheated (by 10-20 deg C).

In all boiler systems, evaporating surfaces for pressures above 14 MPa are almost exclusively arranged in the furnace shaft in the form of water walls which absorb radiant heat.

Water walls may be either bare-tube in design, with small spacings (4-6 mm) left between parallel tubes (Fig. 17.1a), or gas-tight, made from pressed or rolled finned tubes, smooth tubes with rectangular fins welded to them or from smooth tubes with intertubular spacings filled with building-up metal (Fig. 17.1b-d). Water walls in which tubes are welded together and form an all-welded gas-tight structure are alternately called membrane walls, or membranes.

In boiler furnaces which burn low-volatile fuels, the zone of stable ignition and intensive combustion of the fuel is formed at the burner belt level by making water walls from studded tubes and coating them with a refractory material (refractory-faced water walls, Fig. 17.2).

17.2. Reliable Designs of Water Walls

Bare-tube water walls are employed in all hoiler systems with vacuum (balanced draft) furnaces. In naturalcirculation boilers, water walls are almost exclusively arranged vertically or, sometimes, at a steep angle. In once-through and multiple forced circulation boilers, where the motion of the steam-water mixture can be effected at velocities preventing the disturbance of hydraulic conditions, the evaporating heating surfaces can be oriented at any angle, i.e. the water walls may be vertical, horizontal or ascending-descending.

Because of the differences in natural circulation and forced circulation, the reliable designs of water walls for gravity-circulation boilers and oncethrough boilers will be discussed separately.

Methods for increasing circulation reliability. In properly designed and manufactured circulation circuits operating under normal conditions, there are usually no difficulties as regards the reliability of circulation. Let us recall, however, that the driving circulating head decreases with increasing pressure (see Sec. 12.1). An increase in the unit steam-generating capacity of a boiler is associated with increasing the width of the water walls of circulation circuits, which may involve large non-uniformities in the heating of parallel tubes and adversely affect the circulation. With an increasing steam-generating capacity, the heating intensity of evaporating tubes increases substantially.

As power engineering develops and employs increasingly more powerful boiler units for high steam parameters, the reliability requirements of boilers, especially of their circulation circuits, become more and more rigorous.

Disturbances in circulation conditions mainly occur due to uneven heating across the width of a circuit. Nonuniform heating along the height the tubes in a circuit is of minor importance, since in that case all parallel or vertical tubes receive roughly the same quantity of heat and are cooled similarly by the passing water. Uneven heating across the width of a circuit may be caused by an improperly designed circulation circuit (see Fig. 12.11) or by inadequate operating conditions (see Fig. 12.12). The former problem can usually be remedied to some extent at the design stage. Disturbances due to inadequate operating conditions are less determinate. The main cause of non-uniform heating is slagging of tubes. Slagging is never uniform all over the surface of a water wall and depends on many factors, in particular, on the air conditions in the furnace, the distribution of fuel by burners in the furnace, etc. Heavily slagged, and therefore, poorly heated tubes acquire on the whole substantially less heat than do clean tubes and for that reason develop lower driving circulating head, meaning that less cooling (circulating) water passes through them. Such tubes are cooled poorly; in exposed portions free from slag they may become overheated.

With an increase in pressure, especially at p > 16 MPa, the useful circulating head noticeably decreases (see Fig. 12.3). This involves a decrease in the circulation ratio which has a rather strong effect on the temperature conditions in the metal of the heated tubes. For this reason, it is essential to ensure the required circulation ratio at the design stage of circulation circuits.

The reliability of circulation can be increased by increasing the circulation ratio or by dividing wide water walls into narrower sections (water wall sectionalizing).

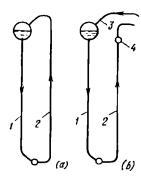


Fig. 17.3. Diagrams of natural-circulation circuits

(a) with direct delivery of a steam-water mixture to boiler drum; (b) with a steam-water mixture passing through a header; 1—water-feeding (downtake) tubes; 2—evaporating (uptake) tubes; 3—steam-freedating tubes; 4—header

Increasing the circulation ratio. In circulation circuits, evaporating tubes can be connected either directly to the boiler drum or through headers and steam-circulating tubes.

For a given steam-generating capacity of a circuit, the required circulation ratio is ensured if the flow rate of water is sufficient for the reliable cooling of the heated tubes. This is achieved by properly selecting the cross section of water-supply and steam-circulating tubes in the circulation circuit (Fig. 17.3).

As shown in Fig. 17.4, for a circuit with steam-generating tubes connected directly to the boiler drum (Fig. 17.3a) and with a small cross section of the downtake tubes, the curve

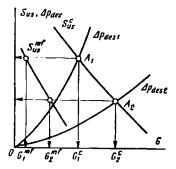


Fig. 17.4. Effect of the resistance of waterfeeding tubes on the flow rate through evaporating tubes

of hydraulic resistance of these tubes, Δp_{desi} , passes quite steeply and intersects the curve of useful heads in the circuit S_{us} at point A_1 (the working point of the circulation diagram). At that point, the hydraulic resistance is too high, and therefore, the circulation velocity and flow rate of water are limited. Under such conditions, the stagnation margin of circulation diminishes sharply, while the limited flow rate of water may fail to ensure reliable heat removal from the evaporating tubes.

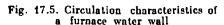
By increasing the cross section of the downtake tubes, their hydraulic resistance is decreased (Δp_{dex2}), thus decreasing the circulation head (working point A_2) and increasing the margin to stagnation. In this way, the total flow rate of water increases through both the uptake tubes and, more important, through the poorly heated tubes, improving their operating conditions.

The cross section required for downtake tubes is determined by calculating the circulation; at high pressures, it may be taken equal to 0.4-0.5 of the cross section of uptake tubes for water walls on furnace sides and 0.7-0.9 of their cross section for platen walls.

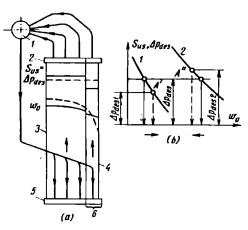
In the circulation circuit with steamcirculating tubes shown in Fig. 17.3b, it is essential, for more reliable circulation, to additionally diminish the hydraulic resistance of the steam-circulating tubes, which can be achieved by increasing their cross section or decreasing their length. The total cross-sectional area of steam-circulating tubes is usually 30-60% that of the steam-generating tubes.

Water wall sectionalizing. Since uneven heating of evaporating tubes is the principal cause of dangerous situations, water walls are sectionalized, i.e. a group of tubes which are heated similarly are combined into a section which is fed separately with water.

Figure 17.5 shows the distribution of velocities in a water wall heated unevenly across its width. The corner tubes receive substantially less heat than those in the middle of the wall and therefore cannot develop the same useful head $S_{us} = \Delta p_{des}$ as in the properly heated tubes in the middle of the wall, so that circulation stagnation, free water level or circulation reversal are likely to appear in them. By providing partitions in the upper and lower header, corner tubes can be separated into an individual circulation circuit (section), so that the circulation velocities in the two circuits can be roughly equalized. The useful head in the separated section will then decrease somewhat, but the margin to stagnation and circulation reversal will increase for the same non-uniformity



(a) distribution of useful head and circulation velocity across the width of a sectionalized water wall: I—drum; 2—upper header; 3—middle water wall tubes; 4—corner water wall tubes; 5—lower header; 5—tubes connected into a separate circuit; — — unsectionalized water wall; — sectionalized water wall; (b) circulation diagram: I— S_{US} of a weakly heated circuit; 2— S_{US} of an intensively heated circuit; Δp_{dess} —pressure gradient before sectionalizing; Δp_{dess} and Δp_{dess} —pressure gradients in the circuits after sectionalizing



of heating across the wall width, and the circulation velocity will be increased (working point A' in Fig. 17.5b). In the section with intensively heated tubes, the useful head will increase and the circulation velocity will decrease somewhat, but the margin to stagnation and circulation reversal in that section still remains sufficiently large (working point A'' in Fig. 17.5b).

In gravity-circulation boilers, furnace water walls are usually made from bare tubes covering all the sides of the furnace. In high- and superhighpressure boilers, tubes with an inside diameter of 40-50 mm are used. Downtake tubes have a diameter of 60-160 mm or more. Large-diameter tubes (600-800 mm) are sometimes em-

ployed in the downtake system of high-capacity boilers.

Figure 17.6 shows a typical lay-out of furnace water walls and their elements in a high-pressure boiler. Water walls 1, 6, and 7 represent a system of parallel vortical tubes. Some portions of the tubes may be steeply inclined to match the shape of the furnace (tubes forming the dry-bottom hopper 9, tubes around burner ports 8, etc.). In high-pressure boiler plants where the available radiant heat released in the furnace is greater than that required for steam generation, evaporating heating surfaces do not cover the entire surface of the furnace. Therefore, other types of heating surfaces can be arranged in it. Should this be the case, evaporating

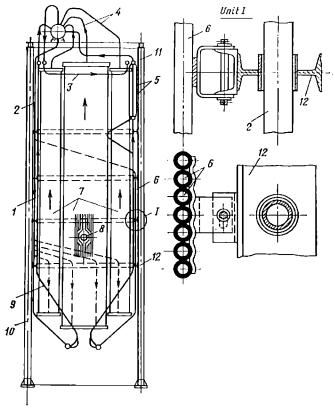


Fig. 17.6. Arrangement of water walls in the furnace of a high-pressure boiler 1—front water wall; 2—downtake tubes; 5—roof water wall; 4—steam-circulating tubes; 5—slaj screen; 5—rear waterjwall; 7—side water wall; 8—tube arrangement around hurner ports; 9—dry hottom*hopper; 10—holler structure; 11—slag screen header; 12—reinforcement belt

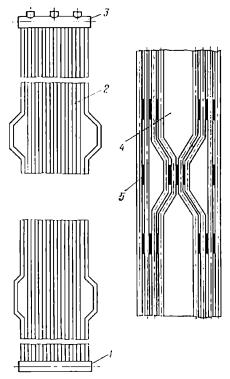


Fig. 17.7. Section of a curtain wall (platen) 1—lower header; 2—tubes; 3—upper header; 4—windows; 5—points of tube welding

surfaces are arranged on the vertical walls of the furnace and superheater surface 3, on the furnace roof. Radiation superheaters can also be arranged at the top of the front wall or along the entire height of the furnace front; in some cases, superheater and evaporating sections are arranged alternately.

All upper headers are mounted roughly on the same height and suspended from the boiler structure. Furnace water walls are provided with rigidity belts 12 of steel sections, which, on heating, can move freely together with tubes. Furnace water walls often carry a boiler setting which is covered on the outside by a casing, or shell (see Sec. 21.1). Thus, the boiler structure carries the suspended water wall system together with the boiler setting and casing. A connection

unit of the ascending or descending tubes to the movable rigidity belts isshown separately in Fig. 17.6 (unit I). The entire water wall tube system and boiler setting attached to it can expand freely downwards.

In high-capacity boilers, curtain walls, or platens, are often used which separate the furnace into a number of vertical shafts (see Sec. 7.1). Platens are suspended in the same manner as side water walls. Some tubes in a platen are bent to form windows in the top portion or along the whole height of it; these are provided to equalize the pressure on both sides of a platen and prevent bending of tubes should a 'puff' occur in the furnace (Fig. 17.7).

Methods for enhancing the reliability of water walls in once-through boilers. Water walls in once-through boilers are arranged either vertically or horizontally (see Fig. 11.1). Depending on the orientation of the tubes, they may receive different quantities of heat along the furnace height and depth and across the width.

In boilers with horizontal or slightly inclined tubes, evaporating tubes are multiply wound around the periphery of the furnace (Fig. 17.8). If one of the furnace sides is heated more intensively than others, this non-uniformity across the width or depth of the furnace (or, what is the same, along the path of the working fluid) will have no effect, since all tubes pass around all the sides of furnace and non-uniform heating will be of the same degree in all parallel tubes.

In order to minimize the mass of the tube system, the furnace water walls of high-capacity high-pressure boilers are made from tubes of a small inside diameter (25-40 mm). In this case, the number of parallel tubes must be increased to maintain the same mass flow rate. These two circumstances, i.e. an increased boiler capacity and smaller tube diameter, require that the width of the tube band in the furnace be increased. With a wider tube band, non-unifor-

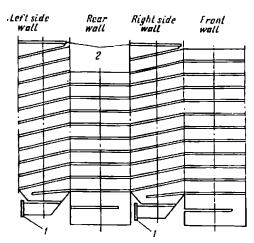


Fig. 17.8. Development of furnace water walls of a Ramzin once-through holler 1-supply headers; 2-discharge headers

mity of heating along the height of the furnace in the parallel tubes of a band becomes more substantial. Because of this, the working fluid flow in high-capacity boilers is divided into a number of parallel flows, i.e. a number of parallel tube bands are arranged in the furnace. Thus, the width of a band and non-uniformity of heating in it can be decreased in proportion to the number of flows (Fig. 17.9). Bands are wound continuously in a helical pattern onto all four sides of the furnace, thus providing for an ascending motion of the flow. Various systems of tube winding in the furnace are in use. Usually, the inclined tubes of a band are arranged on two side walls, while the front and rear walls of the furnace carry horizontal tubes (Fig. 17.8). In other versions, a band rises on the rear side only or else on all four walls. The angle of incline of a band depends on its width (the number and diameter of tubes) and on the width of the furnace walls and is usually equal to 12-15°. The number of tubes in a band is determined by the boiler capacity and the mass velocity of fluid in the tubes. which for the zone of intensive heataken equal to roughly ting is 1000 kg/(m² s) at $p \le 10$ MPa and 1 500-2 000 kg/(m² s) at higher pressures. An excessively high mass velocity may involve a high hydraulic resistance.

The bending of straight portions of horizontal tubes is prevented by fastening them in three points: fixed-

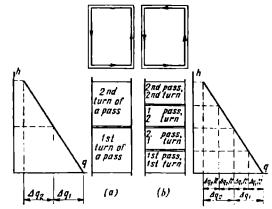


Fig. 17.9. Effect of the horizontal coiling band width on heating non-uniformity

·(a) single-pass coiling (b) two-pass coiling

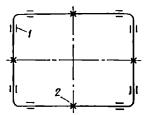


Fig. 17.10. Diagram of fastening of water walls with horizontally coiled tubes t—movable fastenings; 2-fixed fastenings

ly in the middle and in movable supports to allow for thermal expansion, at the ends (Fig. 17.10). Horizontal winding of the tubes simplifies the connection of water walls on adjacent furnace sides, remedies the problem of shadowing the corner tube portions, and improves the conditions of uniform heating of individual tubes, since tube bands extend around the whole perimeter of the furnace. With horizontal tube winding, however, it is more difficult to make water walls in the form of blocks and the tube system has more welded joints. A version of horizontal winding is the meander system, i.e. a system consisting of a number of vertically arranged tube coils

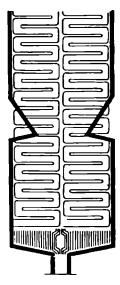


Fig. 17.11. Section of a water wall with horizontal ascending coils

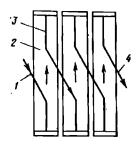


Fig. 17.12. Series connection of sections of a furnace water wall

1—inlet of the working fluid into the water wall;
2—water wall sections; 3—downtake tubes; 4—to next section

with essentially long horizontal portions and short vertical bends (Fig. 17.11). This system is less sensitive to uneven heating, especially when the cross-sectional dimensions of the furnaces are large, and is employed where the water walls are not made gas-tight. Tube fastening in the meander system is, however, more intricate.

Vertical water walls in once-through boilers resemble those in gravity circulation boilers and occupy the whole surface of furnace walls. For more uniform heating, water walls are sectionalized across the furnace width, the sections being connected in series, thus forming a multi-pass system (Fig. 17.12). Individual sections are connected by downtake tubes, which eliminate the maldistribution of heat due to flow intermixing, but make the design and operation of water walls more complicated.

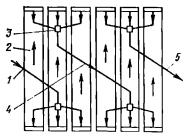


Fig. 17.13. Series connection of parallelconnected sections

t—Inlet of the working fluid; 2: sections; 3—mixer: 4 downtake tubes; 5—to next section

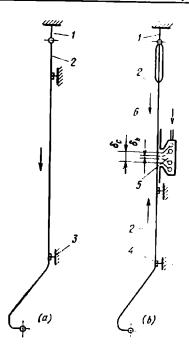


Fig. 17.14. Diagrams of vertical water walls (a) without parting joint; (b) with parting joint; 1—suspension; 2—water wall; 1—movable fastening; 4—support fastening; 5—parting joint seal; 6—air blowing

Water walls of a narrow width are heated essentially uniformly. In high-capacity boilers, narrow vertical water walls are combined into parallel blocks which are connected with one another in series (Fig. 17.13).

Vertical water walls can be conveniently made in the form of blocks, i.e. systems of vertical tubes connected to headers at the ends. Fastenings of walls and moans for their thermal expansion may be various (Fig. 17.14). The tube system of a water wall is usually suspended by its upper header and can expand downwards freely (Fig. 17.14a). In high-capacity boilers, water walls are usually sectionalized vertically, i.e. they have a parting joint amid their height (see Figs. 22.4 and 22.5). In such a system (Fig. 17.14b), the upper section tubes of a water wall are suspended by the upper header from the boiler structure. The lower section tubes are fastened at the

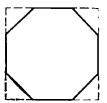


Fig. 17.15. Diagram of octahedral furnace (plan view)

dry-bottom bend to the structure. Both sections can expand towards the parting joint, which has a width of 200-300 mm in the cold state so that, being heated, they come close together to the minimum gap δ_h . Ash deposits in the joint are periodically removed by air blowing.

In order to diminish non-uniform heating across the width of vertical water walls, the corners of the furnace are sometimes 'cut off' to give an octahedral shape in plan (Fig. 17.15). This furnace shape is not very convenient in design, but is advantageous in the respect that a vertical whirl of flame can be easily formed in it if corner burners are arranged tangentially.

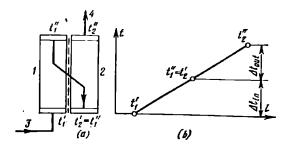
The hydraulic system of boilers is often made in the form of U-, II-, or N-shaped multi-pass sections with vertical uptake and downtake tubes (see Sec. 11.3). Since all parallel tubes in the system pass through the same unevenly heated zones along the furnace height, they receive essentially the same quantity of heat.

17.3. Gas-tight Water Walls and Methods for Enhancing Their Reliability

Water walls in gas-tight boilers are made in the form of vertical sections. To ensure proper strength, the temperature difference between the joined tubes should not exceed 50-100 deg C. This requirement can be easily satisfied in single-pass vertical motion (short tubes) where the temperature difference of the working fluid in adjacent tubes does not exceed the allowable limit and no special measures.

Fig. 17.16. Diagram of an all-welded two-pass water wall (a) and curve of temperature gradients in connected sections at the inlet Δt_{in} and exit Δt_{out} (b).

1 and 2—water wall sections; J and 4—inlet and outlet of the working fluid



are needed. In a two-pass scheme, vertical water wall sections are arranged side by side, with the working fluid flowing successively from one section to another (Fig. 17.16a). A large temperature difference between adjacent sections ($\Delta t > 50\text{-}100 \deg C$) may lead to excessive temperature stresses and even break-through of tubes.

Furnace water walls, especially in supercritical-pressure boilers, operate under heavy conditions: high temperature and pressure of the working fluid, high temperature of the flame, intensive heating, and corrosive flue gases. In this respect, it is extremely important to increase the reliability of the water walls. With proper organization of the processes at both sides of the heat-transfer surface of water walls in gas-tight boilers, this is achieved by decreasing the temperature difference Δt as much as possible. The main methods for decreasing the temperature difference between welded sections are as follows: recirculating the combustion products and the working fluid, intermixing the working fluid along its path (along the length of water walls), and bypassing a portion of the cold flow.

Cold flow bypassing. In this scheme

(Fig. 17.17a), a portion of the working fluid is bypassed around the first section of the heated water walls, which increases the heat absorption per unit flow rate and increases the fluid temperature at the outlet from that section. The temperature difference Δt_{out} between the welded sections is thus reduced and can be retained at the allowable value even in the most critical zone at the outlet from the sections (Fig. 17.17b). The redistribution of the flow rate between a section and a bypass line has almost no effect on Δt_{in} .

In the above scheme, the mass velocity along the path increases: it is the lowest in the first pass of the lower radiation section and the highest in the second pass, which makes it possible to raise the temperature of the fluid at the inlet to and exit from the first pass so that it approaches the respective values in the second pass (Fig. 17.17b). The quantity of by-passed fluid is determined by the ratio of mass velocities of the fluid and the heat release rate in the furnace. It is roughly equal to 20% of D_r , which ensures satisfactory temperature conditions and reliable operation of the water walls.

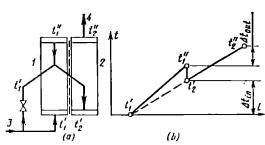


Fig. 17.17. Diagram of a bypassed all-welded two-pass water wall (a) and curve of temperature gradients of connected sections at the inlet Δt_{ln} and exit Δt_{out} (b)

Designations the same as in Fig. 17.16

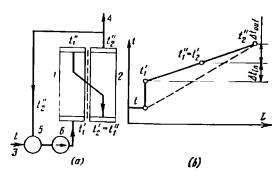


Fig. 17.18. Diagram of an all-welded two-pass water wall with recirculating pump

Designations as in Fig. 17.16; additionally: 5-mixer; 6-recirculating pump

Recirculation of the working fluid. In this method, the flow rate through heavily heated water walls is increased by supplying a portion of the flow which has passed through the lower radiation section (Fig. 17.18a). For the same heating intensity, this method decreases the gain in enthalpy of the working fluid. The temperature of the fluid at the inlet to the heating surface increases, while the temperature at its outlet remains the same as without recirculation (Fig. 17.18b). Accordingly, the temperature difference Δt between welded water walls decreases to a safe value.

Provision of an additional flow of the working fluid at low loads, say at start-up, ensures reliable cooling of the water walls, which is especially important in the combustion of fuel oil, i.e. at a high heat release rate in the furnace. This method makes it possible to reduce the start-up load down to 15% of D_r .

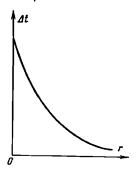


Fig. 17.19. Effect of the working fluid recirculation on temperature gradient in connected sections of an all-welded two-pass water wall

The temperature conditions of adjacent water walls depend on the circulation ratio r which is understood as the ratio of the working fluid flow rate, with recirculation G_r , to the straight flow rate G (without recirculation):

$$r = G_r/G$$

As the recirculation ratio r increases, the temperature difference between adjacent welded sections decreases (Fig. 17.19). Partial ejection of the hot flow (Fig. 17.20) is a version of the recirculation method. Partial ejection decreases the heat absorption in adjacent water wall sections and accordingly decreases the temperature difference Δt along their whole height. Since the flow rate of the working fluid increases and the heat absorption per unit flow rate decreases, maldistribution of heat in water wall tubes be-

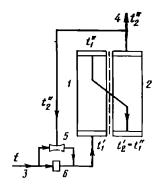
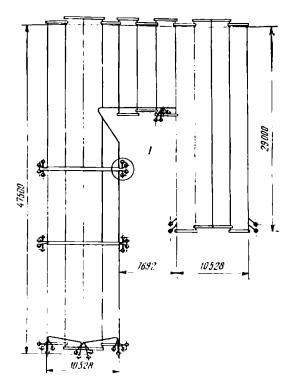


Fig. 17.20. Diagram of an all-welded twopass water wall with ejector (see the curve of temperature gradients in connected sections in Fig. 17.18b)

Designations as in Fig. 17.10; additionally: 5—ejector; 6—throttle

Fig. 17.21. Diagram of water walls of a gas-tight boiler with two horizontal parting joints (see unit I in Fig. 17.22)



comes less probable. The method has, however, certain disadvantages: the elevated hydraulic resistance of the working fluid path and the limited capacity of the ejector.

Intermixing of the working fluid. In intensively heated tubes of furnace water walls, any substantial maldistribution of heat may be dangerous. The risk increases with increasing heat absorption of tubes, i.e. with an increain the tube length. Maldistribution of heat should be minimized where possible, especially in gas-tight boilers. For this reason, water walls of high-capacity boilers (both gas-tight and untight) are divided vertically into stages by parting joints and connected with one another by mixing headers (Fig. 17.21). For better reliability and gas-tightness, the number of parting joints should be as low as possible; for instance, a single joint may be provided between the lower and medium radiation section or between the medium and upper radiation section.

Separation into stages decreases the heat absorption Δi of the fluid in each stage, and therefore, decreases the highest temperature of the tube walls.

The recirculation of combustion products is an effective method for increasing the reliability of water walls (see also Sec. 18.4). Combustion products are taken off at a temperature near 350°C behind the economizer and recycled into the zone of the highest heat release in the furnace. Thus, recirculated gases dilute the oxidant in the zone and prolong the process of combustion, which leads to a lower temperature in the furnace and a lower heat release rate. This is of special importance in fuel oil-fired boilers where the water walls are heated quite intensively.

Gas-tight all-welded water walls are a kind of heating surface for intensified heating. Their mass per unit area of radiant heat-absorbing surface is 10-15% lower than that of hare-tube water walls. The tube pitch cap

be increased by decreasing the number of tubes and selecting their total crosssectional area so as to ensure the required mass flow rate of the working fluid. All-welded water walls operate under more favourable conditions than bare-tube walls, since part of the heat absorbed by tube fins is spread through the metal to the rear sides of tubes and these portions become active heating surfaces. Further, tubes of gas-tight all-welded water walls are held perfectly in-line and thus cannot shadow one another. Gas-tight all-welded water walls may have only a thin layer of heat insulation instead of heavy boiler setting and can be washed by water without the risk of moistening the heat-insulating layer and causing corrosion in inaccessible places.

For a given tube pitch, the reliability of gas-tight all-welded water walls depends on the heating intensity. The allowable intensity of heating is determined by thermal calculation of a water wall which determines the best relationship between the geometrical characteristics (relative pitch, thickness of fins) of a water wall at a given tube diameter and given heating intensity to ensure reliable heat transfer from the fins through the tube to the working fluid (see Sec. 10.4).

For gas-tight all-welded water walls, the uniform heating of their tubes is more essential. This requirement is satisfied better in gas-tight boilers with all-welded vertical water walls and ascending motion of the working fluid. The number of parallel tubes in a wall must be limited to ensure the required mass flow rate of the working fluid. In high-capacity boilers where the furnace shaft has a substantial perimeter, this necessitates an increase in the number of separate flows or of the number of passes connected in series. The former measure is inefficient since it sharply increases the number of tube fittings, complicates automatic control, and impairs reliability of operation. This, in turn, increases the temperature dif-

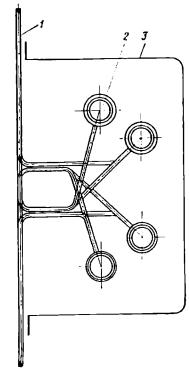


Fig. 17.22. Parting unit of the 'glove' type -furnace water walls: ?--header; J--box

ference between joined tubes and decreases their reliability, and further, increases the number and mass of unheated tubes and thus the hydraulic resistance of the circuit.

In order to decrease the furnace perimeter, gas-tight boilers are designed for an elevated steam-generating capacity of the furnace front (80-120 t/(h m)). The depth of the furnace is increased somewhat to make the furnace almost square in plan, i.e. to obtain the minimal length of its perimeter at a given heat release rate per unit area, q_F .

Figure 17.21 schematically shows the gas-tight all-welded vertical water wall sections of a gas-fuel oil-fired boiler type TGMP-204. They have two parting joints along the height. The 'glove'-type parting joint is shown in Fig. 17.22. It is tightened by a shaped plate and arranged inside a

steel box passing around the whole perimeter of the furnace shaft.

The roof wall is assembled of gastight section units. Its tubes are bent in certain places to form holes for the passage of the tubes of platens and convective tube banks through the roof. These passages are scaled, say, by means of bellows as shown in Fig. 17.23.

Of special importance is the tightness of supercharged boilers where the surplus pressure of combustion products in the furnace is quite high. Ensuring that the roof in such boilers is tight presents certain difficulties,

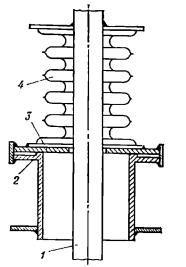


Fig. 17.23. Sealing unit for the passage of tubes through furnace roof 1—tube: 2—fixed support; 3—movable support; 4—bellows

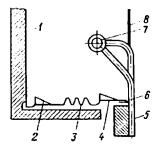


Fig. 17.24. Peripheral sealing of a *hot box*
1—vertical wall of 'hot box'; 8, 5, 4—expansion
compensators; 5—tubes of gas-tight water wall;
6—teeth; 7—header; 8—water wall suspension

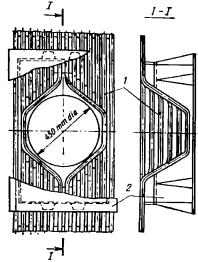


Fig. 17.25. Tube arrangement in an allwelded water wall

I-water wall tubes; 2-welded-to steel frame

since a large number of heating surface tubes pass through the roof. To correct this, a second roof is provided above the main one, so that a 'hot box' is formed between them. Since the hot box and the adjacent side walls expand differently on heating, it is sealed along its periphery. The design of the sealing is shown in Fig. 17.24.

The outside enclosure of a boiler has 450 mm manholes for servicing the gas path. The tubes around a manhole are bent aside and welded to a steel frame (Fig. 17.25). In water walls

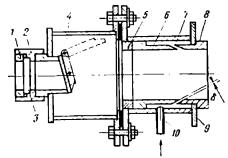


Fig. 17.26. Viewhole of a gas-tight boiler 1—pressure nut; 2—glass holder; J—port glass; 4—hinged flap; 5—check nut; 6—nut to control the gap 0; 7—annular slit for air; 8—come end-plece; 9—Irame; 10—pressurized air in

made from finned tubes, the bent portions around the manholes are replaced by smooth tubes of the same diameter.

The boiler enclosure also has 100 mm viewholes for observing the combustion process and inspecting the state of the heating surfaces. The viewholes are protected by refractory glass. In gas-tight boilers, compressed air is supplied into the viewholes to protect the operator when replacing the glass or introducing measuring instruments into the gas path (Fig. 17.26).

17.4. Refractory-faced Water Walls

Refractory-faced side water walls (Fig. 17.2a and b) and platens (Fig. 17.2c) are mounted in the zones of intensive combustion, such as slagging-bottom furnaces, cyclone furnaces, and the igniting belts of furnaces for low-volatile fuels.

For making a refractory-faced water wall, studs (pins) 10-12 mm in diameter, and 15-20 mm high are welded by resistance welding to the surface of the tubes. The studs form the frame for carbide-silica refractory paste and remove heat from it to the cooled tubes. The refractory facing decreases the heat absorption of the tubes to a small fraction. On the other hand, its conductivity should be sufficiently high to transfer the absorbed heat.

Heat absorption is also associated with the density of the stud arrangement, which is understood as the sum of the cross-sectional area of study per unit area (1 m2) of water wall tubes. At a higher studding density, the refractory facing is held in place more firmly. The limit is determined by the technological possibilities of stud welding. For high-capacity boilers, the density of study is usually 0.15-0.25. The facing is compacted by a pneumatic hammer to ensure its good contact with the surface of studs and tubes. To illustrate this, Fig. 17.27 shows temperature distribution across a refractory-faced smooth-tube water wall (p = 10 MPa).

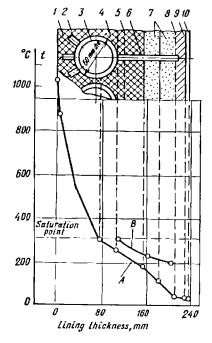


Fig. 17.27. Temperature distribution in a refractory-faced water wall

I—chromite; 2—studs; J—water wall tube; 4—insulation paste; 5, 8—wire net; 6—fastening pln; 7—zolonite plates; 9—tightening plaster; 10—gastight coating; A—temperature of facing; B—temperature of pin δ

Studded water walls operate under heavy temperature conditions, often resulting in burn-off of the facing and studs proper. The service life of refractory-faced water walls depends on a number of factors: furnace temperature, the geometry and material of the studs, the contact resistance between the metal and refractory facing, their conductivities, and the properties of slag. Under comparable conditions. the reliability of refractory-faced water walls can be noticeably increased by shortening the length of the studs to 10-15 mm and increasing the coefficient of thermal conductivity λ above 6 W/(m K).

The height of the stude and the thickness of facing are 'self-controlled', as it were, in the course of operation according to the thermal condi-

tions in the boiler. If the heat transfer through the water wall tubes is too low, the surface facing the furnace is fused together with exposed studs, dissolves in molten slag, and is filled by the slag. As a result, the thickness of the facing and the height of the studs diminish to the best value corresponding to the heat balance of the wall. The carbide-silica facing is sometimes covered on the fireside by a layer of corundum paste which has a higher slag resistance. Double-layer

facing improves the temperature conditions of studs. Refractory-faced studded tubes cause more trouble in maintenance than do smooth tubes when burned-through portions of water walls must be replaced.

In some cases, tubes are used in water walls, on which outer ribs are formed by knurling. Knurled tubes are extremely durable, firmly retain the refractory facing, and are more convenient in repairs.



STEAM SUPERHEATERS AND SUPERHEAT CONTROL

18.1. Classification of Superheaters

Saturated steam produced in evaporating heating surfaces is further superheated to a specified temperature in steam superheaters. The superheater is one of the most critical elements of a hoiler plant, since it is arranged in the high-temperature zone and the steam temperature in it acquires the highest value.

Depending on the method of heat absorption, superheaters are divided into convective superheaters, which are located in the convective gas duct and absorb heat mainly by convection, and radiant superheaters which are arranged on the furnace walls and absorb the radiant heat of the flame. In addition, there is a group of semiradiant platen superheaters that are mounted at the top of the boiler furnace and, partially, in the horizontal gas duct between the radiant and convective heating surfaces.

One must distinguish between primary superheaters which superheat the saturated steam just produced in the boiler, and intermediate (or reheat) superheaters, or simply reheaters, in which the worked-off steam returned from the turbine is superheated once more.

Convective superheaters are made from steel tubes of an internal diameter of 20-30 mm. In reheaters, the diameter of the tubes may be larger, up to 50 mm.

Superheaters are usually made from smooth tubes, as these are less expensive and simpler to manufacture than finned ones. Smooth tubes are less prone to fouling by ash and can be cleaned more easily. A drawback of the smooth-tube heating surfaces is that they have a limited heat absorption at moderate velocities of gas flow. Heat transfer through tubular surfaces is restricted by the heat passing through the external surface of a tube, and therefore, can be increased by extending that surface. Thus, it has been proposed to make superheaters from externally finned tubes, which may have either longitudinal fins (finned tubes proper, Fig. 18.1a) or

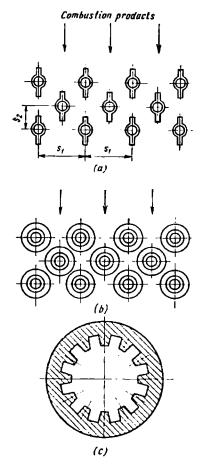


Fig. 18.1. Superheater tube banks
(a) from finned tubes; (b) from cross-finned (gliled) tubes; (c) cross section of a rifled tube

transverse fins (cross-finned, or gilled tubes, Fig. 18.1b).

In high-capacity monobloc units, steam reheating is employed widely. Since the pressure of reheated steam is not high (3-4 MPa), the hydraulic resistance of a reheater should also not be high (the pressure loss not more than 0.2-0.3 MPa). This restricts the mass flow rate of steam and requires the use of tubes of a substantially larger diameter, resulting in a lower heattransfer coefficient on the internal wall. Low values of the heat transfer coefficient α_2 at an intensive heating of the external surfaces of a reheater

can sometimes lead to an inadmissible rise of temperature, especially at the reheater outlet, which may turn out to be too high for the pearlitic steel of which the reheater is made and necessitate a change to a more expensive austenitic steel. The temperature of the reheater walls can be diminished by arranging te reheater in a zone of moderate heating; this, however, will require a significant increase in its surface and may be economically unfavourable. The internal heat transfer in the outlet ('hot') section of a reheater can be intensified by using internally finned, or rifled, tubes (Fig. 18.1c). This measure develops the internal surface and substantially decreases the wall temperature.

Superheater tubes are bent into coils with a bending radius of not less than $1.9d_{ex}$. The coil ends are welded to round-section headers. Tube coils may have one or more passes (Fig. 18.2a-d). Multi-pass coils are employed in high-capacity boilers. With multi-pass coils, there may be not enough place on a header for welding the tube ends. Furthermore, the header walls may be weakened. This is avoided in a 'glove'-type connection of tubes to headers (Fig. 18.2e).

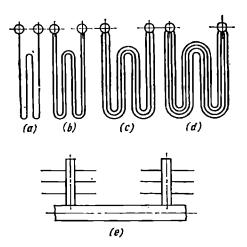
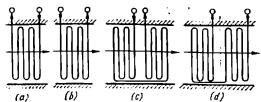


Fig. 18.2. Types of superheater coils
(a) single-pass; (b) two-pass; (c) three-pass; (d) four-pass; (e) glove-type multi-pass

Fig. 18.3. Motion of steam and combustion products in convective superheaters

(a) counter-flow; (b) parallel flow; (c) and (d) mixed flow



Depending on the direction of flows of steam and combustion products, superheaters may be of parallel-flow, counter-flow or mixed type (Fig. 18.3).

A counter-flow superheater (Fig. 18.3a) develops the highest temperature gradient between combustion products and steam, which makes it possible to diminish the heating surface area and the use of metal. A drawback of this scheme is the elevated hazard

of burn-through of the last coils in the steam path, since both the steam and the combustion products have the highest temperature here and the tube metal is under heavy temperature conditions. In a parallel-flow superheater (Fig. 18.3b), the temperature gradient is much lower and the tube metal operates under more favourable temperature conditions, since the tube coils in which the steam has the highest temperature are in contact with the

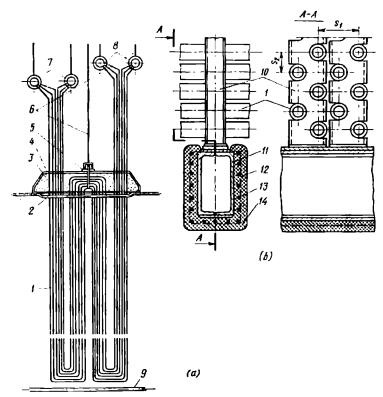


Fig. 18.4. Tube fastenings of (a) vertical and (b) horizontal superheaters

1—superheater tubes; 2—roof water wall; 3—refractory packing; 4—sealing box; 3—clamp; 6—suspension; 7 and 8—supply and discharge headers; 9—water wall on the bottom of horizontal duct; 10—stands;

11—support heam; 12—metallic mesh; 13—reinforcing bars; 14—heat insulation

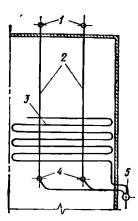


Fig. 18.5. Attachment of a horizontal convective superheater to water wall tubes I and 4—headers of suspended tubes; 3—superheater bank; 5—supply header of superheater

combustion products which have already been partially cooled in the inlet section of the superheater. The mixed-flow scheme (Fig. 18.3c and d) offers the best solution as regards the cost and reliability of convective superheaters.

The tube coils of superheaters are arranged either vertically or horizontally. Vertical superheaters have a simpler design, can be fastened more reliably, and are less subject to slagging, but are undrainable, i.e. the condensate cannot be drained off from them completely. This can cause internal corrosion and involve certain difficulties in boiler firing. Horizontal

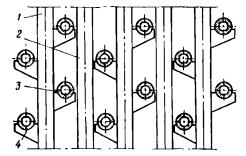


Fig. 18.6. Attachment of tubes of a horizontal superheater bank to suspended tubes

1—suspended tubes; 2—spacing tubes; 3—superheater tubes; 4—supporting strips

superheaters have a more intricate design as regards their fastening, but enable complete drainage of the condensate.

Tube coils can be fastened by various methods depending on their arrangement. In vertical superheaters. the top loops are held by clamps. The headers and clamps are suspended from the boiler structure (Fig. 18.4a). Horizontal superheaters operating at the temperature of combustion products up to 700°C are fastened to stands made from heat-resisting steel sheets (Fig. 18.4b). At higher temperatures, the stands may be subject to intensive high-temperature gas corrosion, especially in gas and fuel oil fired boilers. In this case, horizontal banks are suspended from the tubes of the steamwater path of the boiler (Fig. 18.5). The supporting tubes are of the same diameter as other tubes in the heating surface, but have supporting strips welded to them (Fig. 18.6).

Radiant superheaters. If the heating surface of a radiant superheater in drum-type boilers is not large, it can be located on the furnace roof (Fig. 17.6, item 3). Otherwise, it can occupy part of the vertical walls of the furnace. In once-through boilers, the radiant superheater usually occupies the furnace roof, the upper and medium radiation sections, and the walls of the horizontal flue duct.

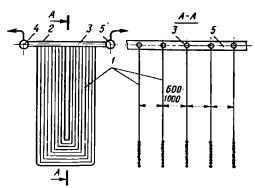
The tubes of vertical and horizontal radiant superheaters are fastened in the same way as in evaporating heating surfaces. They should be allowed to expand freely on heating.

Radiant superheaters have certain advantages, in particular, a low hydraulic resistance (fractions of a megapascal) and a low resistance on the gas side, since they do not block up the gas duct.

Platen superheaters. A platen superheater is usually a system of tubes assembled into a flat gas-tight band with inlet and outlet headers. Platens are arranged vertically or horizontally with spacings of 600-1 000 mm between them. Vertical platens are faste-

Fig. 18.7. A vertical platen superheater

1—platens; 2—platen supply header; 3 platen discharge header; 4—inlet hox of platen superheater; 5—outlet box of platen superheater



ned by the upper header (Fig. 18.7). The principal advantage of platens is that they absorb both radiant and convective heat, which ensures their high thermal efficiency with a low resistance in the gas path. Platen superheaters can absorb up to 50% of the total heat of superheating. Softened ash particles continuously stick to the surface

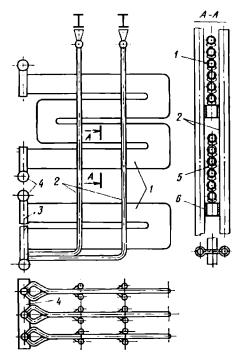


Fig. 18.8. Horizontal platen superheater mounted on cooled suspended tubes 1—horizontal platens; 2—suspended cooled tubes; 3—headers; 4—collecting boxes; 5—spacers; 6—support strip of platen bank

of platens and solidify on their tubes. Those, however, are self-cleaned due to vibration, so that ash deposits cannot grow to an appreciable thickness. Dense deposits can only form in boilers fired on highly-slagging fuel grades. A drawback of vertical platens with a top header position is that they cannot be drained off. Horizontal platens are fastened by certain tubes from the platen bank (Fig. 18.8).

Tubes in a platen may widely differ in length and shape. Parallel tubes may absorb different quantities of heat. Front tubes are heated much more strongly than other tubes in a platen. As a result, the external tubes of a platen may turn out to operate under critical conditions. The reliability of platens is enhanced by making one or more tubes that operate under the heaviest conditions from a more heat-resisting grade of steel or by using tubes of a larger diameter (Fig. 18.9a); by making the external tubes of a shorter length (Fig. 18.9b); or by shadowing the external tubes by a frame of tubes of another heating

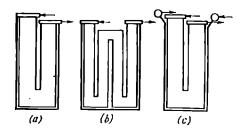


Fig. 18.9. Methods of protection of extreme tubes in platen superheaters

surface operating at a lower temperature (Fig. 18.9c).

Platens are usually made from smooth tubes. At some power stations, membrane-type platens with finned tubes have been tested. They are slagged less heavily and can be cleaned more easily; their tubes are held more firmly in line. Horizontal membranes can be supported only at the ends, i.e. without intermediate supports or suspensions, since they have a sufficiently rigid structure.

18.2. Operation and Reliability of Superheaters

As regards conditions for creep and scaling, the metal of superheaters operates practically at the upper safe limit. The margin of temperature rise of the superheater metal is rather limited, and therefore, the allowable maldistribution of heat (see Sec. 11.4) and the allowable temperature rise above the average (rated) value are very low.

On the other hand, in high-capacity boilers with a high steam superheat, and therefore, high heat absorption of the superheater, an actual maldistribution of heat can easily exceed the allowable limit, which will lead to a loss in reliability of the superheater operation. Up to recent times, the reliability of heating surfaces, in particular of superheaters, was calculated for the service life of tube metal of 100 000 hours (or roughly 15 years). Testing and operation of monobloc units have proven that the service life of metal can be increased almost twofold. On the other hand, an increase in the temperature of the superheater by 15-20 deg C above the rated value can shorten the service life by roughly 50%.

In high-capacity boilers which have a large cross section of gas ducts, the superheaters are heated extremely unevenly over the width and height of horizontal ducts and over the width and depth of vertical ducts. For vertical superheaters with tube coils ar-

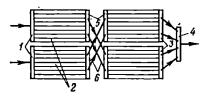


Fig. 18.10. Steam transfer by means of cross-over tubes

1—supply beader; 2—heating surface; 3—discharge header; 4—steam-collecting box; 5—intermediate header; 6—cross-over tubes

ranged in the plane of motion of combustion products, and where steam repeatedly changes the direction of motion from ascending to descending and vice versa, non-uniformity of heating over the height of the gas duct is inessential since all parallel coils are under the same heating conditions (provided that heating is uniform over the duct width). In horizontal superheaters with tube coils lying in the plane of motion of combustion products and with steam changing repeatedly the direction of motion along the duct and in the opposite direction, i.e. with the coils arranged perpendicular to the front of gases, non-uniform heating along the length of the duct is also inessential. On the other hand, heating in both types of superheater may be substantially non-uniform, especially in boilers with a wide furnace front.

The effect of non-uniform heating across the width of a gas duct can be diminished by sectionalizing the superheater across its width and along the length of the duct. In such a case, it is essential to transfer the steam flow in a cross-over manner, i.e. from one section to the opposite end of a next section. Steam transfer between sections can be effected by means of tubes (Fig. 18.10) or headers (Fig. 18.11).

In high-capacity boilers with a wide furnace front, steam transfer between superheater sections presents a much more serious problem and involves an increase in the hydraulic resistance of the superheater. In such cases it is advisable to organize combustion in the furnace so that non-uni-

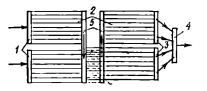


Fig. 18.11. Steam transfer by means of headers

Designations as in Fig. 18.10

form heating of the gas ducts over their width is minimized.

The major portion of the superheater in a boiler is usually arranged in the horizontal gas duct and immedia tely above it, i.e. at the entry to the convective furnace shaft. In high-capacity boilers, the depth of the shaft and the height of the horizontal duct are roughly of the same substantial size, which at high velocities of combustion products results in a highly non-uniform velocity and concentration fields, especially of coarse ash particles at turns and at the entry to the convective shaft (see Fig. 16.5). When tube coils of the superheater are arranged perpendicular to the gas front, all coils are subjected to wear by ash, which increases the scope of superheater repairs. In the superheaters with tube coils parallel to the gas front, intensive wear is concentrated on a small group of tubes at the rear wall of the gas duct.

The cooling conditions for the tubes of superheaters and reheaters are different. Superheaters are cooled by steam from the very moment the boiler is fired and can be arranged directly in the furnace, as well as in the convective duct. Reheaters begin to receive steam only when the turbine is started up, i.e. they are not cooled at all for an appreciably long time during boiler start-up. The same is true in an emergency shut-down of the boiler. In order to avoid metal superheating, reheaters are preferably of the convective type and, less frequently, of the platen type, and are arranged in the zones of moderate heating where the temperature of combustion products is not higher than 850°C. In a number

of circuits, reheaters are cooled at start-up and shut-down by fresh steamsupplied through an attemperator.

18.3. Positioning of Superheaters

The temperature of tube metal in a superheater is substantially higher than that of the steam that flows in it. If there is no maldistribution of heat between the parallel tubes of a superheater, the relationship between these two temperatures is given by formula (10.12). Under comparable conditions (i.e. with the same values of t_a, α_2, δ_w , and λ_w), a radiant superheater is heated much more intensively $(q \approx 500-600 \text{ kW/m}^2)$ than a convective superheater which operates in a zone of moderate temperatures of comhustion products and has a heating intensity which is an order of magnitude lower than the radiant superheater. In view of the maldistribution of heat, the wall temperature of the worst malfunctioning tubes of a radiant superheater may exceed the temperature of the steam in it by 100-150 deg C.

Positioning of the superheater in the combustion product flow depends on steam parameters (see Table 17.1) and the arrangement of evaporating heating surfaces. For instance, in medium-pressure boilers (p = 4 MPa, $t_{ss} = 440$ °C) where the heat absorption by the superheater does not exceed 20% of the total absorption of the boiler and the evaporating surfaces occupy the entire surface of the furnace, the superheater is of the convective type and positioned immediately behind a membrane wall (Fig. 18.12a). Tubes in the front rows of the superheater coils are spaced widely apart to avoid slagging. The superheater is made by the combined schemein order to protect the metal in the last coils from the effect of excessively high temperatures. It has vertical tubes and is undrainable.

In boilers for a pressure of 10 MPa and $t_{ee} = 540^{\circ}$ C (Fig. 18.12b), the superheater consists of a convective section and a radiant platen section.

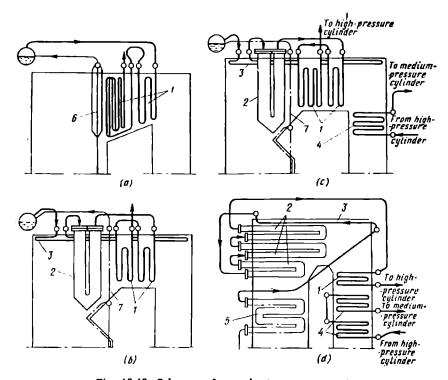


Fig. 18.12. Schemes of superheater arrangement \cdot (a) p=4 MPa, $t_{ss}=440^{\circ}$ C; (b) p=10 MPa, $t_{ss}=540^{\circ}$ C; (c) p=14 MPa, $t_{ss}=545^{\circ}/545^{\circ}$ C; (d) p=25.5 MPa, $t_{ss}=545^{\circ}/545^{\circ}$ C; 1—convective superheater; 2—platen superheater; 3—roof superheater: 4—convective reheater; 5—furnace water walls; 6—slag screen; 7—suspended tulves

The platen section is located in the upper portion of the furnace in front of the suspended tubes of the rear water wall and the convective section is in the horizontal portion of the convective gas duct. The platens are the first to meet the combustion products and thus protect the convective section from slagging by ash. The two sections of the superheater are connected in series and the hot bank is arranged in the convective gas duct. Steam is fed from the boiler drum first into the small radiant section and further into the roof water wall, the platen section and the convective section.

In boilers for pressures of 14 MPa and $t_{ss} = 545/545^{\circ}$ C, there are usually a main superheater and an intermediate (reheat) superheater (reheater), as shown in Fig. 18.12c. The arrangement of the main superheater is

essentially the same as in boilers for p=10 MPa and $t_{ss}=540$ °C. The reheater is positioned in the convective furnace shaft in the zone of moderate temperatures of combustion products (below 850°C).

With a further increase in steam parameters, the fraction of heat to be spent on superheating considerably increases, so that the superheater must be partially positioned in the boiler furnace. In this case, it may consist of radiant, platen and convective sections.

The scheme for connection of the elements of a superheater should take into consideration that the radiant section operates under heavier conditions than the convective section. For this reason, the radiant section is commonly connected to be the first in the steam path. Platen-type superheaters

are also rather sonsitive to disturbances in the furnace, especially those arranged directly in the furnace. A typical scheme for connection of superheater elements in supercritical-pressure boilers is shown in Fig. 18.12d. Upon passing the medium and upper radiation sections of the boiler, steam directed into a roof water wall and then into the platens and the convective bank of the superheater. The reheater is made in the form of two convective banks. Both the superheater and the reheater are drainable.

18.4. Superheat Temperature Control

The control characteristic, i.e. the relationship between the boiler load and the temperature of superheated steam, is different for various superheaters. A typical feature of radiant superheaters is that the temperature of superheated steam decreases with an increase in the boiler load (curve I in Fig. 18.13). The quantity of heat absorbed by radiant heating surfaces depends mainly on the theoretical temperature of fuel combustion, the flame emissivity, and the thermal efficiency of the water walls (see Sec. 20.3). These parameters, however, depend only very slightly on the quantity of burned fuel, i.e. on the boiler load. For this reason, the heat absorption in a radiant superheater increases less quickly than the flow rate of steam through it, and therefore, the unit heat absorption (per unit of steam flow

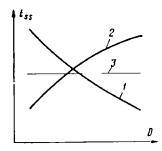


Fig. 18.13. Control characteristics of superheaters

I—radiant; 2—convective; J—combined

rate) decreases. In a convective superheater, the quantity of combustion products passing through it increases almost in proportion to the boiler load, with the result that the heat transfer by conduction increases in proportion to the power of 0.6-0.65 of gas velocity. Thus, as the direct heat absorption in the furnace decreases and the temporature of the combustion products at the furnace exit increases, the temperature gradient in the zone of convective superheater becomes higher. These two circumstances are responsible for the fact that the temperature of superheated steam increases more quickly than does the boiler load (curve 2 in Fig. 18.13).

With properly selected dimensions of the radiant and convective superheater sections, it would be possible to obtain a constant temperature of superheated steam (curve 3 in Fig. 18.13). Under practical conditions, however, the temperature of superheated steam will vary due to certain operating factors, which include the temperature of feed water, the excess air ratio in the furnace, slagging of the furnace water walls, especially of the superheater, and the moisture content of the fuel.

In drum-type boilers where the superheater area is fixed, the steam-generating capacity decreases as the temperature of feed water decreases. With a constant fuel consumption. this means that the quantity of heat of the gases absorbed in the superheater zone per unit flow rate of steam increases, resulting in a higher temperature of superheated steam. In once-through boilers, on the other hand, a lower feed water temperature leads to a respective drop in the temperature of superheated steam, since the unit heat absorption remains constant. An increase in the excess air ratio in the furnace is associated with an increased quantity of combustion products passing through the convective superheater, resulting in more intensive heat exchange in this section and a higher temperature of superheated steam. An increased moisture content of fuel leads to a higher temperature of superheated steam, since the quantity of combustion products passing through the superheater increases and the combustion products contain more triatomic gases; therefore, they have a higher emissivity and increase the heat-transfer coefficient on the fireside. Slagging of the evaporating water walls results in a higher temperature of combustion products at the outlet from the furnace and in a respective increase in steam temperature. By contrast, slagging of the superheater proper impairs heat exchange in it and decreases the temperature of the superheated steam.

In once-through boilers where the zone of phase transition is indefinite, the actual surface area of the superheater section varies with variations in operating factors owing to displacement of the end of evaporation zone. In this case, the temperature of superheated steam can be maintained constant by controlling the ratio between the water flow rate and fuel consumption. On the other hand, once-through boilers have a low accumulating capacity (i.e. a low bulk of water and metal), and are thus rather sensitive to variations in the water flow rate and fuel consumption, which result in variations in the temperature of superheated steam.

In operation, the temperature of superheated steam is allowed to vary from +5 to -10 deg C of the rated value. Even combined radiant-convective superheaters cannot ensure a constant steam temperature in operation within these limits, because of which steam boilers are provided with means for superheated steam temperature control. Since the load of a power station may vary considerably, the control means should maintain the rated superheat temperature within wide limits of steam-generating capacities. The temperature of superheated steam should be kept at the rated value in the load range from 30% to 100% of the rated value and that of reheated steam, from 60% to 100%.

Methods of superheated steam temperature control. There are two main methods for controlling the temperature of superheated steam: steam control and gas control.

Steam control is based on reducing the enthalpy of steam by transferring part of its heat to feed water or by injecting demineralized water into steam. These methods are usually employed for controlling the temperature of live steam. Steam control is also used for controlling the temperature of reheated steam, but in this case it is based on the redistribution of heat between the fresh and reheated steam.

Gas control is based on varying the heat absorption on the fireside of heating surfaces to a value ensuring the desired level of superheated steam temperature. These methods include recirculation of combustion products, bypassing part of the combustion products around the superheater, and variation of the position of the flame in the furnace. Gas control is employed mainly for controlling the temperature of reheated steam or, if there is no reheating of the temperature of live steam.

Steam control. Steam control has found wide application and is effected in heat exchangers: surface-type and spray-type attemperators, or desuperheaters. In this case, the surface area of the superheater has a certain reserve, and the excessive superheat is removed in the attemperator. Surface- and spray-type attemperators are used to control the temperature of main, or live, steam. The temperature of reheated steam is controlled in live-steam reheaters. Injection of condensate into the flow of reheated steam is economically unfavourable, since the additional quantity of superheated steamthus formed enters together with the main steam into the turbine bypassing the high-pressure cylinder.

The attemperator (desuperheater) can be mounted behind the superhea-

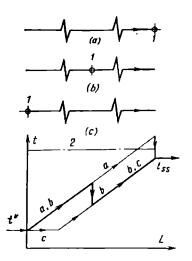


Fig. 18.14. Effect of the position of attemperator on superheated steam temperature in the superheater path

(a) positioned downstream of the superheater; (b) inserted between superheater sections; (c) upstream of the superheater; t-attemperator; 2-allowable steam temperature

ter, between the superheater sections, or in the saturated-steam line. With the attemperator arranged at the outlet of the superheater (Fig. 18.14a), the latter remains unprotected against excessive rises in the steam temperature; for this reason, the method is not used for controlling the temperature of main superheated steam.

In other versions of the attemperator arrangements, the turbine and superheater are properly protected. The scheme with the attemperator inserted between the superheater sections has however a lower inertia, especially at high pressures (Fig. 18.14b). This method of steam temperature control shortens the steam path behind the controller and the time for varying the quantity of heat accumulated in the metal of the superhenter section behind the controller. The delaved effect of the attemperator on the temperature of superheated steam is determined by the heat absorption of the path behind the attemperator. With a lower heat absorption of that path, the desired temperature is attained more quickly and control is more flexible. The drawback of this arrangement (Fig. 18.14c) is that it lengthens the time of response.

A surface-type attemperator is essentially a tubular heat exchanger with cooling water (usually feed water) flowing in the tubes and the flow of steam to be cooled (or condensed, if the attemperator is mounted at the superheater inlet), around the tubes (Fig. 18.15). The tube system consists of U-shaped coils mounted in a cylindrical housing.

A surface-type attemperator is connected in series with the boiler economizer (Fig. 18.16). At any load of the boiler, all feed water passes through the economizer, thus ensuring its reliable cooling. Surface-type attemperators are employed in low-capacity boilers.

Water injection is the simplest method of steam control. A spray-type attemperator (Fig. 18.17) is essentially a portion of a header in which condensate is injected into the flow of superheated steam. It is introduced through an atomizing nozzle with a number of

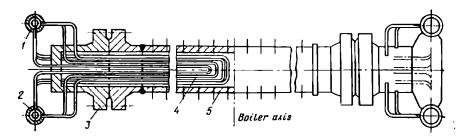


Fig. 18.15. Surface-type attemperator 1 and 8—supply and discharge headers; 3—cover; 4—coils; 5—box

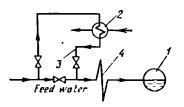


Fig. 18.16. Connection of a surface attemperator into the circuit

1—drum; 2—attemperator; 3—water drainage from attemperator; 4—economizer

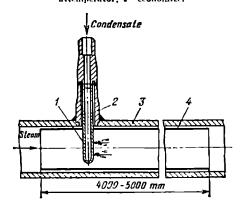


Fig. 18.17. Sprny-type attemperator 1—atomizing nozzle; 2—pipe connection; 3—header; 4—protective Jacket

holes 3-6 mm in diameter. The nozzle is covered by a jacket (with a 6-10 mm gap between them) to avoid contact between the relatively cold condensate jets and the header walls which are at the same temperature as the superheated steam. The length of the jacket (4-5 m) ensures complete evaporation of the water droplets inside the jacket.

Spray-type attemperators are sensitive to the quality of spray water. Once-through boilers and many drum-

type boilers are fed with the turbine condensate and are often provided with spray-type attemperators for condensate spraying. If the feed water for a drum-type boiler is mineralized, condensate for spraying is obtained directly from the boiler (Fig. 18.18) by condensing saturated steam.

It should be noted that if the spraytype attemperator is arranged too close to the superheater outlet, the steam temperature before the spray nozzle may be too high and will impair the operating conditions of the header metal in that place. In some circuits, two or even three injection points are provided in the steam path (Fig. 18.19a and b) in order to control the temperature of steam and prevent excessive temperatures behind the superheater sections. The last injection point in a circuit is arranged before the last stage of the superheater at $\Delta i = 160-300 \text{ kJ/kg}.$

Live-steam reheater. If the main superheater is of the radiant type and the reheater, convective, a drop in the boiler load will increase the temperature of main (live) steam and decrease that of reheated steam (see Fig. 18.13). To equalize the temperature in both, it is reasonable to take some of the heat from the live steam and transfer it to the reheated steam. This may be done in a live-steam reheater, which is used essentially for redistributing the heat absorbed by radiation and convection.

A section of a live-steam roheater contains 10-20 tubes that are 25-35 mm in diameter, mounted in a 300-400 mm header (Fig. 18.20). The apparatus is U-shaped for better compensation of thermal expansion of the

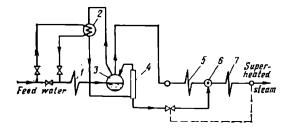


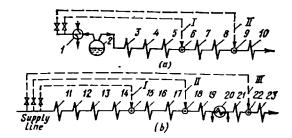
Fig. 18.18. Superheat control by condensate spraying

J-economizer; 2-condenser; J-drum;

d-condensate tank: 5 and 7-superheater; d-spray-type attemperator

Fig. 18.19. Superheat control by (a) two and (b) three spray-type attemperators

1—condenser; 2—drum; 3—radiant superheater on furnace wall; 4 and 18—radiant roof superheater; 5, 7, 17 and 19—platens; 6, 9, 15, 18 and 82—spray-type attemperators; 8, 10, 21 and 31—conomizer; 12—lower sections; 11—economizer; 12—lower radiation section; 13—medium radiation section; 14—upper radiation section; 20—live-steam reheater (see Fig. 18.20); 1, 11 and 111—spray lines



tube system and more compactness. Live (superheated) steam moves in the tubes and reheated steam in the housing (header). The temperature is controlled by bypassing part of the flow around the heat exchanger.

Live-steam reheaters usually have a number of sections, up to a few tens in high-capacity plants. The sections are connected in parallel. The range of steam temperature control is 30-40 deg C.

A live-steam reheater absorbs the heat of the superheated steam and is located outside the gas path. Other reheating surfaces in the boiler are arranged in the convective gas duct and connected in series.

The main connection schemes for a live-steam reheater in the circuit are shown in Fig. 18.21. In all of them, the live-steam reheater is connected

behind all elements having radiation characteristics (see Fig. 18.13, curve 1). Accordingly, the heat absorption by the live-steam reheater increases with decreasing boiler load.

In the circuit shown in Fig. 18.21a, all superheated steam passes through the live-steam reheater. The temperature of reheated steam is controlled by varying its flow rate, i.e. by directing part of the reheated steam flow around the live-steam reheater through a steam bypass valve, which changes the coefficient of heat transfer from the wall to steam, α_2 , and the temperature gradient in the live-steam reheater. The drawback with this circuit is a high power loss due to an appreciable hydraulic resistance in the superheated steam path.

In the circuit illustrated in Fig. 18.21b, the live-steam reheater is vir-

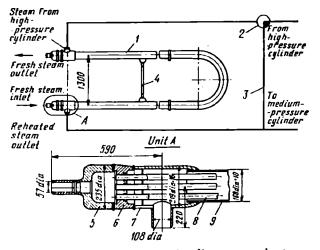


Fig. 18.20. Typical section of a live-steam reheater

1—live-steam reheater section; 2—control valve; 3—by-pass line; 4—spacer strip; 5—bottom; 6—section head; 7—sealing disc; 8—heat-exchange tubes; 9—section housing

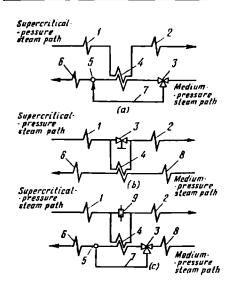


Fig. 18.21. Connection of live-steam reheaters into the circuit

1—radiant and semi-radiant superheater sections; 2—convective superheater sections; 3—steam bypass valve; 4—live-steam reheater; 5—mixing chamber; 6 and 8—convective reheater sections; 7 bypass line; 9—orifice plate

tually not used at the rated load of the boiler. Almost all the superheated steam passes through the bypass valve and only 5% of it enters the livesteam heater. At a reduced load, the flow rate of the reheated steam through the live-steam reheater decreases, and the fraction of superheated steam passing through it is increased by closing the bypass valve. This ensures a constant reheated steam temporature at reduced loads of the unit. The drawback of this scheme is that the control valves must have a large cross section and operate at high or supercritical steam parameters.

The scheme shown in Fig. 18.21c is a compromise between the former two. In this case, an uncontrollable fraction of live steam flows through the live-steam reheater; this is established at the rated boiler load and remains constant at other loads.

The flow of reheated steam is usually divided into the main flow, which passes through a live-steam reheater, and the bypass flow around that rehe-

ater; these flows are usually controlled by means of bypass valves.

Gas control. Gas control is effected by recirculating combustion products, tilting burners, turning on and off the burner tiers, or bypassing combustion products.

Gas control is employed to maintain the desired temperature of the reheated steam, but it is associated with changes in combustion conditions in the furnace, and therefore, can influence the temperature of the superheated steam, too. Gas control involves additional power consumption for draft and heat loss with waste gases and also has an effect on the temperature of the superheated steam, which complicates boiler operation.

Gas control cannot ensure the specified temperature of the superheated steam and for this reason is employed in modern boilers only in combination with steam control. In circuits with steam reheating, the application of the two independent control methods is inevitable.

Recirculation of combustion products. Combustion products at a temperature of 250-350°C are taken off the convective shaft by the recirculation blower and subsequently returned to the furnace. This makes it possible to redistribute the heat between the heating surfaces, depending on the circulation ratio. At a higher recirculation ratio, a greater thermal effect is obtained. The lowest recirculation ratio to avoid a reverse flow in the recirculation circuit and prevent the blower from cooling is roughly 5%.

Recirculated combustion products can be introduced into the upper or lower portion of the furnace (Fig. 18.22). The temperature and aerodynamic conditions in the zone of motion of recirculated gases may differ, depending on the recirculation circuit and recirculation ratio. The introduction of recirculated gases into the lower portion of the furnace decreases the direct heat absorption in the furnace and accordingly increases the temperature of the combustion products

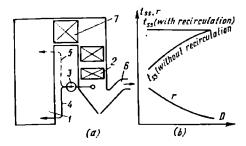


Fig. 18.22. Steam temperature control by recirculation of combustion products (a) recirculation scheme; (b) effect of boiler steam-generating capacity on the superheat temperature and the fraction of recirculated combustion pro-ducts; 1—furnace; 2—economizer; 3—recirculating fan; 4—admission of cold combustion products into the lower portion of furnace; 5—combustion products recirculated into the upper portion of fur-nace; 4—admission of the state of the propersity of the proper

nace; 6-to regenerative air heater; 7-superheater

at the furnace outlet. Recirculation increases the flow rate of gases through the superheater. These two circumstances increase the convective heat exchange, and therefore, the temperature of the superheated steam. Since recirculation decreases the direct heat exchange in the furnace, it has a positive role in protecting the water walls of the lower radiation section against overheating (see Sec. 17.3). Recirculation of combustion products is increased at low loads, when the temperature of the superheated steam drops, and is turned off at high loads, when steam superheating increases; as a result, the volume of combustion products at the outlet from the plant varies only slightly. Since, however, the volume of combustion products at low loads becomes higher, this leads to an increase in ϑ_{wg} and q_{z} and an elevated fuel consumption. It is advisable to introduce recirculated gases into the hot air box of the burners.

Recycling combustion products into the top portion of the furnace has no essential effect on furnace operation, but substantially lowers the temperature of combustion products in the superheater, thus preventing slagging of the superheater tubes, but somewhat decreasing the heat absorption.

Flame control in the furnace. Heat absorption by furnace water walls is determined by the temperature level and the pattern of temperature distribution. Heat absorption in the furnace, and therefore, \$7, can be varied by varying the position of the flame. This, in turn, changes the heat absorption by the reheater which is arranged in the convective gas duct. For instance, if the burners are tilted downwards. the total radiant heat absorption by the water walls, Q_r/Q_f , increases, while the temperature at the furnace outlet, 07, decreases (Fig. 18.23), resulting in a lower heat absorption by the reheater. On the other hand, if the burners are tilted upwards, the heat absorption by the water walls decreases, while the temperature of combustion products at the furnace outlet increases. Thus, as the boiler load drops down, the temperature of the reheated steam diminishes (Fig. 18.13, curve 2) and the burners are tilted upwards so as to raise the steam temperature. Gas control by burner tilting can maintain a constant temperature of reheated steam at boiler loads ranging from $100\,\%$ to $70\,\%$.

The position of the flame can also be changed by turning some of the burner tiers on and off. If the furnace has three tiers of burners and the total fuel consumption by them corresponds to 150% of the steam-generating capacity, operation with any

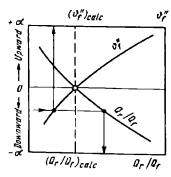


Fig. 18.23. Effect of the angle of burner tilting on the heat absorption of furnace water walls and the temperature of combustion products at furnace exit

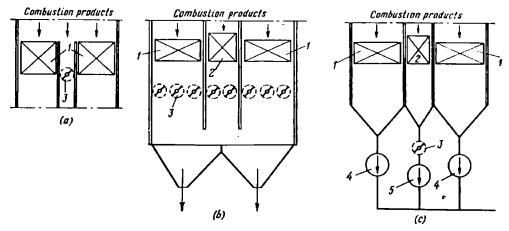


Fig. 18.24. Schemes of temperature control of superheated steam by bypassing of combustion products

(a) passing combustion products through a free gas duct; (b) distribution of combustion products between parallel gas ducts by means of control gate valves; (c) distribution of combustion products by means of a control fan; 1—reheater; 2—economizer; 3—control gate valve; 4—main fan; 5—control fan

two tiers turned on can ensure 100-% load. Thus, at high loads, i.e. when steam superheating increases, the lower tiers are turned on, and at lower loads, the upper tiers are turned on.

Bypassing combustion products. The temperature of reheated steam can be controlled by passing the combustion products around the reheater. Various schemes for effecting the method are illustrated in Fig. 18.24. In Fig. 18.24a, a 'free' gas duct is provided between the reheater sections, and the gas flow rate through it is

controlled by a gate valve. The latter operates under heavy conditions, which explains why this scheme is not popular. In Fig. 18.24b and c, the reheater sections and the economizer are mounted in parallel gas ducts ('split duct') and the gases are distributed between them either by gate valves arranged behind the heating surfaces (as in 'b'), or by exhaust fans (as in 'c'). The latter two methods are more reliable than the first, but complicate the design and increase the cost of the plant.



LOW-TEMPERATURE HEATING SURFACES

19.1. Arrangement of Low-temperature Heating Surfaces

The low-temperature heating surfaces of a boiler include the air heater and the economizer. The design of the-

se elements should pursue the following objects: to intensify heat exchange, minimize the dimensions of the elements with a moderate use of metal, and minimize the effects of abrasion wear by ash, fouling and corrosion.

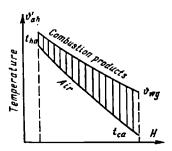


Fig. 19.1. Variation of temperature gradient during air heating

The air heater operates under conditions which differ from those of the economizer or other heating surfaces. It has the lowest temperature gradient between combustion products and air and the lowest heat-transfer coefficient. For this reason, the surface area of the air heater is usually greater than the total area of all elements of the steam-water path, and in high-capacity beilers may range from a few tens to a few hundreds of thousands of square metres.

The air in the air heater is a medium that has a low water vapour content. On the other hand, combustion products which pass through it usually have a high water vapour concentration (depending on the content of moisture and hydrogen in the fuel), as well as of triatomic gases (CO₂ and SO,), and their volume and heat capacity are higher than the respective values for air. The volume of combustion products also increases due to the presence of excess air. As a result, the ratio of the water equivalents of air and combustion products, $\Psi =$ $= c_a V_a/c_g V_g$, in the air heater is always less than unity. This means that air is heated in the air heater more quickly than the combustion products are cooled. For instance, with a low moisture content of the fuel, air is heated in the air heater on the average by 1.2 deg C per each degree the combustion products are cooled, or with fuels of a high moisture content by 1.4 deg C per each degree. Thus, as air is being heated, the temperature gradient, which determines the intensity of heat exchange, decreases to a minimum at the 'hot' end of the air heater (Fig. 19.1). Because of this, the temperature level of waste gases is determined not by the temperature gradient between the fluids at the cold end of the air heater, but by the economically efficient temperature gradient Δt at the hot end, which is lower than the former. This temperature difference is usually not less than 30-40 deg C. Any further increase in the temperature of hot air is inefficient in view of the very weak heat exchange in the hot portion of the air heater (Δt is too low). Thus, an increase in the temperature of hot air is associated with an increase in the temperature of waste gases or an increase in the heating area of the air heater. Figure 19.2 shows the temperatures of waste gases which depend on the temperature of air preheating at various ratios of water equivalents $(\Psi < 1)$, $t_{ca} = 30 \deg C$ and the temperature gradient at the hot end $\Delta t =$ = 40 deg C. As may be seen, to preheat the air to 400-420°C (fuel anthracite) at $\Psi = 0.8$, the temperature of the waste gases at the air heater outlet should be not less than 140-150°C. For fuels with a higher moisture content (brown coal), $\Psi = 0.65$, and the temperature of waste gases for the same level of air preheating must be $\theta_{\omega g} = 200-220^{\circ}\text{C}$, which is economically inefficient. In contrast to this, in a counter-current economizer the temperature gradient at the

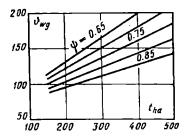


Fig. 19.2. Effect of air heating on the temperature of waste gases at various ratios of water equivalents

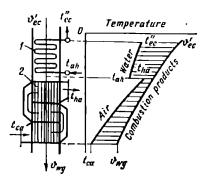


Fig. 19.3. Distribution of temperature gradients in a single-stage arrangement of low-temperature heating surfaces

J-economizer: 2-air heater

hot end increases, since water has an appreciably higher heat capacity.

The low-temperature elements of a boiler can be connected into a single-stage or two-stage circuit.

In a single-stage circuit, the economizer and air heater are arranged in series in the gas path and connected by the counter-flow scheme (Fig. 19.3). This scheme limits the possibility of air heating in the air heater. In the single-stage circuit, it is essential to properly select the gas temperature at the boundary between the economizer and air heater. Noting the ratio of water equivalents, the most efficient temperature of waste gases is obtained on air preheating to 250-350°C. If air should be preheated to 350-450°C, the air heater is of the two-stage type and the economizer is arranged between its sections (Fig. 19.4). The essence of the two-stage circuit is an increased temperature gradient Δt at the hot end of the air heater, since its hotter (second) stage is in the zone where combustion products have a higher temperature. This makes it possible to hold the temperature of waste gases at an appreciably low level.

Air heaters are made of carbon steel for which the allowable temperature is not more than 500°C; with an air preheating temperature of 420°C, this means that the temperature of the combustion products should be not higher than 580°C. The temperature of the combustion products behind the superheater is usually higher, 600-650°C, so that a hot section of the economizer is mounted before the second stage of the air heater in order to protect the metal in the latter. The two-stage circuit of the air heater and economizer substantially increases the height of the convective shaft and the cost of construction; that is why it is employed only with fuels for which a high preheating of air is essential.

19.2. Economizers

Plain-tube economizers. A continuous loop-type economizer is the main type of economizer employed in boilers for various pressures. In order to intensify heat transfer and minimize fouling, the economizer coils are

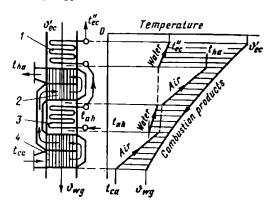


Fig. 19.4. Distribution of temperature gradients in a two-stage arrangement of low-temperature heating surfaces

t and 3—second and first stage of the economizer; 2 and 4—second and first stage of the air beaters

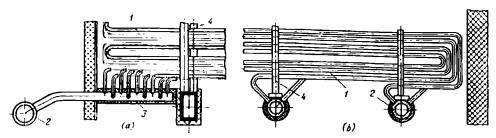


Fig. 19.5. Connection of economizer coils to the header

(a) by means of connection pipes passing through the boiler setting; (b) with the headers arranged in the gas duct; 1—coils; 2—header; 3—connection pipes; 4—coil-supporting structure

made from steel tubes of a small diameter (internal diameter 20-30 mm and wall thickness 2.5-3.5 mm). As in other heating surfaces, the ends of the coils are connected to headers which are arranged beyond the zone of gas heating. The coils are connected to the headers by welding. In highcapacity boilers where the number of economizer coils is large, requiring that many tubes be passed through the refractory lining for connection to an external header, thus increasing air inleakage, the coil ends are connected to a small number of intermediate pipe connections, as shown in Fig. 19.5a. In gas-tight boilers, the economizer headers are almost always mounted in the gas duct and serve as economizer supports (Fig. 19.5b). A staggered arrangement is preferable for the economizer tubes. Motion of water in an economizer is ascending to ensure the free exit of gases and steam with water. To facilitate operation and repairs, economizers are usually sectionalized in the path of combustion products, with sections (banks) of a height up to 1 m. The spacings between the sections are usually 550-600 mm wide.

Economizer coils can be arranged perpendicular or parallel to the boiler front. In the former case (Fig. 19.6a), their length is not large and is determined by the depth of the gas duct This simplifies coil fastening, but all coils in a bank are subjected to abrasion wear by ash. With economizer coils arranged parallel to the boiler

front, the length of the supply header and the number of tubes can be significantly decreased but the length of the tubes in the coils is greater and requires a more reliable means of fastening. In boilers with a wide front, two-sided symmetrical economizers are employed with the headers arranged at the two sides (Fig. 19.6b).

Water in the steel tubes of economizers may be vaporized. Accordingly, economizers may be either non-boiling, where water at the outlet is subcooled below the saturation point or boiling, in which a certain quantity of steam forms in the outlet portion. The steam content of water at the economizer outlet should not exceed 25%. Both types of economizer do not differ substantially in design.

Extended heating surfaces of economizers. In order to intensify heat absorption on the gas side and make the apparatus more compact, rectangular fins are often welded to the plain tubes of an economizer (Fig. 19.7a). This increases the use of metal per unit heating surface area, but gives an essential gain in heat absorption, so that, for the same amount of metal

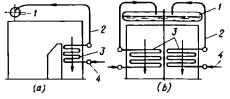


Fig. 19.6. Economizer arrangement 1-drum; 2-water tubes; 3-economizer; 4-supply headers

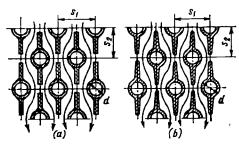


Fig. 19.7. Finned tubes of economizers
(a) welded-on fins; (b) integral fins

and same draft, the volume occupied by a finned heating surface with rectangular fins is 25-30% smaller than that of a surface with plain tubes. Besides, plate steel is cheaper than tubes.

In recent times, profiled finned tubes (with trapezoidal fins) have been adopted for use in economizers (Fig. 19.7b). Finned tubes decrease the dimensions of an economizer by 40-50%.

Cross-finned tubes (with helical or circular fins) are also used in economizers. They can be employed in boilers fired on non-slagging fuels.

Membrane-type heating surfaces may be promising in economizers (Fig. 19.8). They are made from plaintube coils with steel-sheet spacers 2-3 mm thick welded inbetween the straight portions of the tubes. Membrane-type economizers are more efficient than the plain-tube type, consume less metal for the same heat absorption, and are quite reliable in

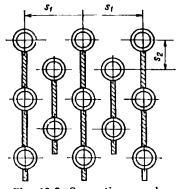


Fig. 19.8. Convective membrane

operation. Economizer membranes may be made with a more extended surface ('lobe-type' membranes).

19.3. Air Heaters

Operating conditions and classification of air heaters. Modern high-capacity boilers are always provided with an air heater. The role of the air heater increases with the unit power of a plant. This is associated with the fact that the temperature of combustion products behind the economizer is still rather high (350-450°C). By utilizing this heat in the air heater, the temperature of combustion products can be lowered to 120-160°C. Air preheating increases boiler efficiency.

On the other hand, the air heater proper operates in the zone of the lowest temperature of combustion products, so that part of its surface (in the cold end) may be at a temperature equal to or lower than the dew point of combustion products. Under such conditions, a moisture film can form in the colder portion and cause corrosion and fouling. These adverse effects are enhanced in the combustion of high-sulphur fuel oils and at high excess air ratios. In gas-tight boilers where fuel can be burned at a low excess air ratio, the corrosion rate is lower (see Sec. 16.3).

Air heaters operate either by the recuperative or the regenerative principle. In recuperative air heaters, heat from combustion products or another heat source is transferred continuously to air through a heating surface. In regenerative air heaters, the heating surface is swept alternately by combustion products and air and is thus alternately heated and cooled.

Air can be heated in air heaters by combustion products either directly (Fig. 19.9a) or through an intermediate heat-transfer agent which may be in the form of solid packing (Fig. 19.9b), hot water (Fig. 19.9d) or bleeder steam from the turbine (Fig. 19.9c and e).

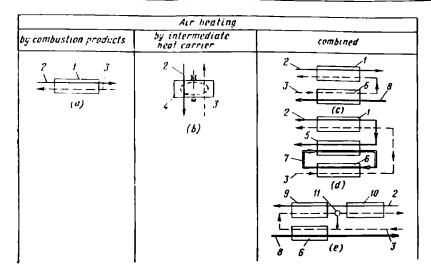


Fig. 19.9. Methods of air heating

(a) tubular air heater; (b) regenerative air heater; (c) air preheating; (d) combination of low-pressure economizer and air heater; (c) cascade air heater; J—tubular air heater; J—combustion products; J—air; 4—regenerative air heater; 5—low-pressure economizer; 6—air heater; 7—water; 8—steam; 9—cascade section of tubular air heater; 10—main tubular air heater; 11—mixer

Direct heating of air by combustion products. Recuperative air heaters are mostly of the tubular type. They are simple in manufacture, but consume much metal and occupy a large space.

Tubular air heaters usually have a vertical tube arrangement (Fig. 19.10). Tubes with an external diameter of 30-40 mm and a wall thickness of 1.2-1.5 mm are welded by the ends to tube plates and arranged in a staggered order. The upper tube plate has a thickness of 20-25 mm and the lower one, 15-20 mm.

Combustion products flow through the tubes (longitudinal flow) and give up their heat to the air which moves between the tubes (cross-flow), i.e. the working fluids are in cross-current flow relative to each other. As is known, the most compact heat exchanger is obtained with a purely counter-current fluid flow, but this is unrealizable in tubular air heaters. The counter-current scheme can be approximated rather closely by causing the air to move in a number of passes between the tubes. The air heater is divided by a number of partitions in the

air path so as to obtain the optimal air velocity for the conditions of heat exchange. Air boxes are provided in places where the air flow changes direction. A tubular air heater has an external steel casing and its lower tube plate bears against a frame which is fastened to the stands of the boiler structure. The tube system can expand upwards on heating. The upper tube plate is connected to the gas duct above it by means of a gas-tight sealing with expansion compensators (Fig. 19.11). For convenience of transport and mounting, tubular air heaters are usually in the form of separate cubical sections.

In boilers of moderate capacity, the tubular air heater is usually of the single-flow scheme with air entering the wider side of the apparatus (Fig. 19.10). In boilers of a higher capacity, i.e. with a larger air flow rate, the single flow scheme requires a greater height of the air heater, resulting in a lower temperature gradient. The height of the apparatus can be substantially diminished by using the split-flow scheme, i.e. by dividing the air

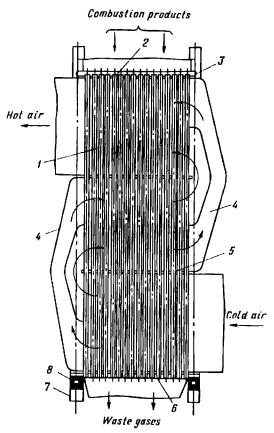


Fig. 19.10. Tubular air heater 1—tubes; 2 and 6—tube plates; 3—expansion compensator; 4—box, 5—partitions; 7—framework; 8—frame

in the heater into two or more separate flows; this makes it possible to increase the number of passes for air,

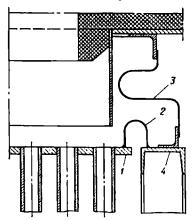


Fig. 19.11. Tube-plate packing

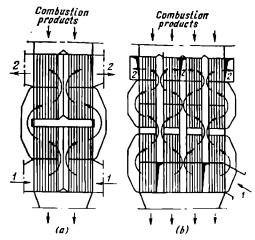
1-tube plate; 2 and 3-expansion compensators;
4-framework beam

and therefore, the temperature gradient (Fig. 19.12a). The combination of the split-flow scheme with densely arranged small-diameter tubes makes the tubular air heater rather compact. In boilers of very high capacities, air in the heater is divided into a greater number of flows (Fig. 19.12b).

With the same velocity of combustion products and air, the coefficient of heat transfer at the gas side of an air heater α_1 (longitudinal flow) is lower than the heat-transfer coefficient at the air side α_2 (cross flow). Heat transfer can be intensified by increasing the coefficient α_1 in longitudinal flow, for instance, by making the heating surfaces from corrugated tubes (Fig. 19.13) of a sinusoidal shape with a constant radius of curvature. To avoid clogging of the tubes by ash, corrugated-tube air heaters are conne-

Fig. 19.12. Air paths in tubular air heaters

1-cold air entry; 2- hot air exit



cted by the reverse scheme, that is air moves in them through the tubes (longitudinal flow) and combustion gases, between the tubes (cross flow).

Corrugated-tube air heaters are more efficient than the straight-tube type and, for the same capacity, their mass may be 50% lower. Though corrugated tubes are more expensive, the cost of a corrugated-tube air heater turns out to be 33% lower than that of the conventional type. Tubular air heaters are simple in design, reliable in operation, and are more tight than other types of air heaters.

Air heating by combustion products through an intermediate heat-transfer agent. From the standpoint of thermodynamics, it does not matter whether heat is transferred from combustion products directly through a wall or first to an intermediate heat-transfer agent and then to air. In the latter case, each of the processes involved can be carried out independently and under optimal conditions.

The principle of air heating through an intermediate heat-transfer medium is realized in regenerative air heaters which are employed widely at thermal power stations. A regenerative air heater has a solid packing—usually a pack of steel sheets which are heated by combustion products and then give up their heat to air. It is typical of air heaters with an intermediate heattransfer agent that corrosion on the heating surfaces has practically no effect on the air inleakage into the gas path of the boiler.

Regenerative air heaters. Regenerative air heaters employed at thermal power stations are mostly of the rotary type. The heat-transfer packing is formed from thin corrugated or plane steel sheets with narrow channels (of an equivalent diameter $d_{eq} =$ = 6-9 mm) formed between them for the passage of combustion products and air (Fig. 19.14). The packing is placed into a hollow cylindrical rotor which is divided by radial partitions into isolated sectors. The rotor (Fig. 19.15) is driven by an electric motor through a gear or a cogwheel transmission and rotates slowly (at a speed of $0.008-0.065 \text{ s}^{-1}$).

The housing of the regenerator is divided into two sections by the upper and lower sector plates with necks for admitting combustion products and air. The gas and air flows move

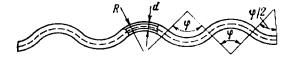


Fig. 19.13. Corrugated tube

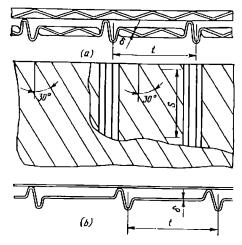


Fig. 19.14. Packing profiles of regenerative air heaters

(a) packing of the bot section with corrugated packing sheet; (b) packing of the cold section

ugh the rotor separately and continuously, while the rotating packing intersects them alternately. The two flows are in counter-current relations. Thus, when some of the sectors are heated by combustion products, others give up the accumulated heat to air.

The packing of a rotary air heater should ensure an intensive heat exchange and have an aerodynamic resistance as low as possible. Its designs are diverse, the most popular being shown in Fig. 19.14. The packing profile is chosen according to the tempe-

rature conditions. Its hotter portion is made so as to intensify heat transfer (as in Fig. 19.14a) and consists of two types of sheets: corrugated sheets and spacer sheets which have wavy ridges oriented at an angle of 30° to the flow direction. The colder portion of packing has a simpler design (Fig. 19.14b) and consists of corrugated sheets and plane sheets between them. Corrugations turbulize the flow and enhance heat transfer.

Since rotary air heaters have channels of non-uniform cross section in the packing, convective heat transfer in them can be intensified more easily than in tubular air heaters. Further, they are simpler in manufacture, since corrugated sheets for the packing can be made from large sheets by rolling or stamping.

The quality of rotor packing may have an essential effect on the coefficient of utilization of the heating surface area. With poor filling of the rotor, combustion products and air may partly encircle the packing, thus decreasing the air heater efficiency.

The heating surface area of 1 m³ of the rotor packing is usually 300-340 m². Contact of the packing with the gas and air flow is limited in time (usually less than 30 s). The rotor packing should be heated quickly in the gas flow and quickly give up its heat to air. Packing sheets of a thickness of only 0.6-1.2 mm can transfer an

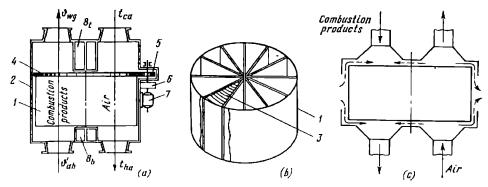


Fig. 19.15. Rotary regenerative air heater
(a) longitudinal section; (b) rotor; (c) points of air pass-over; 1—rotor; 2—stationary housing; 3—packing;
4—driven gear wheel; 5—driving gear wheel; 6—reducer gear; 7—electric drive motor; 8₁ and 8₂—top and bottom sector plate to separate gas and air flow; ——air pass-over

appreciable quantity of heat. Thin sheets are also favourable in that they vibrate in the gas or air flow and thus are cleaned from ash. In some types of regenerative air heaters, the packing is made from metallic, ceramic or glass spheres.

Corrugated-sheet packing has a serious drawback: thin sheets are destroyed quickly by corrosion and abrasion. Another drawback is that the heattransfer coefficient in longitudinal flow is rather low.

Regenerative air heaters are usually made with a rotor of a diameter up to 10 m. or even 15-17 m in high-capacity plants, which rotates on a vertical axis. When the rotor is of a large mass, it requires a heavy-duty radial thrust bearing. With horizontal arrangement of the rotor, the rotor load can be distributed between two ball bearings of a smaller size; fur-

thermore, horizontal-rotor air heaters can be arranged more easily in the gas path of the boiler.

The gas and air flows in elements of a regenerative air heater produce an appreciable pressure gradient, which is roughly the same in balanced-draft and supercharged gas paths and constitutes 7-8 kPa. With a substantial gap between the rotor and stationary structures, this pressure gradient can lead to an overflow of a part of the air to the gas side. Moreover, cold air can be sucked in at the periphery of the air heater gas side and some air can be lost in the air portion (see Fig. 19.15c). The total rated inleakage in regenerative air heaters is around 20%, but may be substantially higher in operation. Gas and air overflow in a rotary air heater is prevented by sealings in the inlet and outlet of the rotor housing: annular sealings at the external and internal circumference of the rotor and a number of radial sealings which separate gas and air sections. The peripheral scaling of a standard rotary air heater is illustrated in Fig. 19.16. It is made in the form of a cut-through sealing ring whose elements form a very narrow

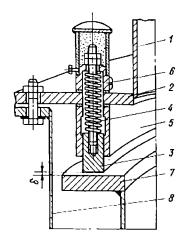


Fig. 19.16. Sealing of regenerative air heater t—wall of gas-supply pipe; 2—cover flange; 3—block; 4—spring; 5—guide plate screwed to flange; 6—cap screwed to flange; 7—rotor flange; 8—cylindrical casing; 6—gap

gap δ with the rotating flange of the rotor. The width of the gap is controlled by springs protected by gas-tight caps. The internal annular and the radial sealings are designed in a similar manner. The quantity of air transferred between the rotating plates into the gas path (and vice versa) in rotary air heaters may be substantial. In some rotary air heater designs, the air that passes through the sealings is sucked off and returned into the air path to the intake of the blow fan.

Regenerative air heaters have found wide application in high-capacity monobloc units. They are more intricate in design than the recuperative type, but occupy substantially less space, have a low aerodynamic resistance and, further, corrosion of the heating surfaces does not increase air inleakage. Regenerative air heaters also have certain drawbacks-for instance, they have rotating elements (the rotor), intricate sealings which separate the gas and air flows, and a high overflow of air into the gas path. The essential drawback of regenerative air heaters with corrugated-sheet packing is that the sheets buckle, thus preventing air heating above 300-350°C.

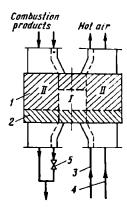


Fig. 19.17. Regenerative air heater with separated air flows

I—hot packing; 2—cold packing; 3—primary air;
 4—secondary air; 5—gate valve; I and II—primary and secondary air sections

In some power plants, for instance, when burning low-volatile (anthracite) or high-moisture fuels, primary air should be heated to a higher temperature than secondary air. In other plants, such as in coal-pulverizing systoms with an intermediate bunker. the aerodynamic resistance of the paths of primary and secondary air may be substantially different. In such cases it is advisable to employ rotary air heaters with separation of the air and gas flow into sections (Fig. 19.17). The apparatus has a separating ring and an additional sealing. The place for mounting the separating ring is determined by the ratio of the required cross-sectional areas for the passage of primary and secondary air. The separating ring separates both the air flow and the gas flow. Air temperature is controlled by gate valves in the gas path, and the air flow rate by gate valves in the air path.

Combined air heating. In addition to conventional heating by low-temperature heat of combustion products, air can be preheated in steam air heaters by low-temperature steam from the regenerative system of the turbine. This method is advantageous for combatting low-temperature corrosion and fouling in the combustion of high-sulphur fuels. The steam air heater

utilizes the latent heat of condensation of worked-off steam from the turbine and thus decreases the heat loss in the circuit. Air can also be preheated by hot water produced in an economizer by utilizing the low-temperature heat of waste gases. Steam air heaters and low-pressure economizers are usually employed for preheating air which is then further heated in conventional air heaters.

In modern high-capacity boilers, the steam air heater is essentially a tubular heat exchanger arranged between the discharge of the blow fan and the inlet stago to the main air heater (Fig. 19.18). Bleeder steam from the turbine, at a temperature of around 120°C, flows in the tubes while air flows around them (cross-current flow).

In winter at substantial sub-zero temperatures of the atmospheric air (below — 15°C), the inlet tubes of a steam air heater and the condensate drainage lines may frost up. Schemes have been developed for preheating air in power units at various climatic conditions.

In the low-pressure economizer-air heater system, air is preheated in the air heater by hot water from the low-pressure economizer. Such systems operate by the closed cycle and preheat air before it is finally heated by combustion products in a tubular air heater (Fig. 19.19). An advantage of the circuit is that it substantially de-

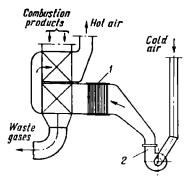


Fig. 19.18. Diagram of air preheating in a steam air heater

1-steam air heater; 2-blower

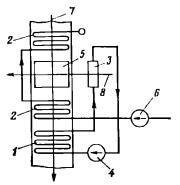


Fig. 19.19. Combined scheme of air preheating with economizer and air heater

I—low-pressure economizer; ?—high-pressure economizer; 3—air heater; 4—circulation pump; 5 air preheater; 6—feed-water pump; 7—combustion products; 8—air

creases air inleakage. The system is made closed in order to prevent contamination of the condensate by corrosion products; the circulation of deaerated water in it is effected by a pump. The temperature of the water is slightly above the dew point, which helps prevent corrosion in the low-pressure economizer.

In a cascade air heater, all cube sections, except for the last one in the gas path, are heated by the total flow rate of the combustion products and cooled by the total flow rate of air. Corrosion in the low-temperature portion of the apparatus is prevented or minimized by maintaining the temperature of the metal above the dew point of the combustion products. To form such conditions, the cube section which is the last in the gas path and the first in the air path is fed with the total flow rate of the combustion pro-

ducts and a low flow rate of air (Fig. 19.20a). Cold air is divided into two flows. The smaller flow (30-40% of the total air flow rate) passes through the air heater and then through the cascade stage; the greater portion of air (70-60% of the total flow rate) is bypassed and mixed with the first portion behind the cascade stage, after which the mixed air moves in a counter-current flow through the hot stages of the cascade air heater.

A simplified temperature curve of a cascade air heater is shown in Fig. 19.20b. As may be seen, air temperature in the preliminary air heater and cascade stage increases more rapidly, since the flow rate of air through them is small. For this reason, these heating surfaces can be made rather compact. Combustion products are cooled less appreciably in the cascade stage, and therefore, their temperature is higher.

19.4. Corrosion Control of Air Heaters

Among all the methods for corrosion control in low-temperature heating surfaces, the most efficient are raising the working temperature of the motal above the dew point, organizing the operation of an air heater on the low-corrosion part of the curve $K = f(t_w)$, as seen in Fig. 16.9, and fuel combustion at the lowest excess air ratio.

Increasing the metal temperature above the dew point $t_{d,p}$, is the most common method for preventing low-temperature gas corrosion. Water vapours are most likely to condense at start-up and low loads of the boiler

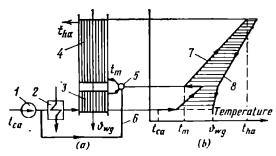


Fig. 19.20. Cascade air heater
(a) connection circuit; (b) temperature
curve; 1-blower; s-air heater; 4main section of air preheater; 4-mixer; 6-bypass line; 7-air; s-combustion products

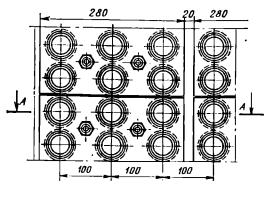
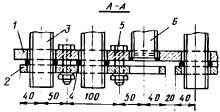


Fig. 19.21. Fastening unit of a glass-tube air heater

1—tube plate; 2—pressure plate; 3-glass
tube; 4--rubber ring; 5—spring washer;
6—steel tube



plant, i.e. at a low temperature of the combustion products. Such conditions occur, however, only for a short time during boiler operation. Besides, at low loads corrosion processes occur substantially less intensively.

The local temperature of the working surface of an air heater is determined from the formula

$$(t_w)_i = (t_a)_i + (q/\alpha_2)_i$$

from which it follows that, for the given heating conditions, the temperature of the wall in the coldest point at the inlet of air to an air heater depends on the inlet air temperature t_a and the coefficient of heat transfer from wall to air, α_2 . Therefore, it can be raised by raising t_a or decreasing α_2 . The latter circumstance, however, contradicts the general tendency to minimize the heating surfaces.

A universally applicable method for preventing gas corrosion is by raising the inlet temperature of air, which is realized by preheating the air in steam air heaters.

With any method employed for raising the inlet temperature of air, it helps to separate the colder portion of the air heater as an individual section in which corrosion wear will be much higher than elsowhere. This simplifies repairs of the air heater, since only the separated section must be replaced. The service life of regenerative air heaters can be increased and repairs made easier by making the packing of the colder section from sheets 1.0-1.5 mm thick (with the sheet thickness in the hotter portion being 0.5-0.8 mm).

In new fuel oil-fired boilers, in the initial heating zone of the air heater where low-temperature corrosion may be especially intensive, steel tubes are replaced by glass tubes with a diameter of 30-40 mm and a wall thickness of approximately 4 mm. A glass-tube air heater has essentially the same design as a conventional air heater with steel tubes, but glass tubes are horizontally staggered. The combustion products pass around a bank of glass tubes and the air moves in the tubes. Glass tubes are fastened in tube plates by means of pressure stool plates and ring washers made of heat-resistant rubber (Fig. 19.21). The structure is made rigid by means of a

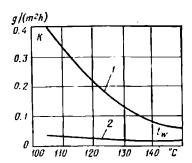


Fig. 19.22. Corrosion rate in cold packing of regenerative air heaters

1-metallic packing: 2-enamelled-metal packing

number of steel tube ties welded between the glass tubes.

The working temperature range of the air heater is from 10° at the inlet to 80-85°C at the outlet. In winter, air is preheated to 10°C in a steam air heater.

In recent times, low-temperature sulphuric corrosion in air heaters is prevented by applying corrosion-resistant coatings to their heating surfaces or by making these from ceramic materials. Acid- and heat-resistant enamels are used for this purpose, in particular, for coating the metallic packing of the colder portion of regenerative air heaters. The thickness of an enamel coating is roughly the same as that of steel sheets (0.5-0.6 mm). The rate of corresion on enamelled heating surfaces is much lower than on bare metal (Fig. 19.22); ash deposits on enamelled surfaces are thinner and can be removed more easily.

Under comparable conditions, low-temperature gas corresion is more intensive at a higher sulphur content of the fuel used. It is especially fast in boiler plants that fire high-sulphur fuel oil.

Combustion of high-sulphur fuel oil at the minimal excess air ratio is an effective means of diminishing low-temperature sulphuric corrosion. With a lower value of α_f and a lower excess of oxygen, combustion products contain less SO₃ (see Sec. 16.3), and therefore, their corrosion activity is lo-

wer. In a particular temperature interval, fly ash particles can stick to heating surfaces and form a moist film on attaining the dew point $t_{d,p,\cdot}$, in which ash particles are cemented into a dense mass. Ash deposits diminish the free cross section for the passage of combustion products and sometimes completely clog some tubes in air heaters.

The operating conditions in the air heater are improved by introducing various additives into high-sulphur fuel oil (alkali additives, an aqueous solution of magnesium chloride, etc.). Additives lower the dew point $t_{d,p}$. of combustion products and neutralize the solution of sulphuric acid that forms on the surface. The deposits on the heating surfaces become loose and can be easily removed by shot-blasting. Further, liquid additives diminish the amount of deposits, improve the conditions of fuel oil combustion, and decrease clogging of the burner nozzles by fuel oil coke. In solid fuel combustion, moderate quantities of additions give no positive results and additions in amounts comparable with the ash content of solid fuel are too complicated and economically unfavourable.

In some plants, air is heated in amounts exceeding the quantity of air required for combustion. The heat of surplus air can then be used for

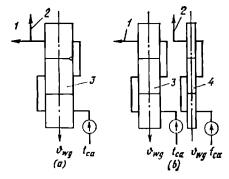


Fig. 19.23. Diagrams of excess air preheating

(o) in the main air heater; (b) in an auxiliary air heater; I—air for fuel combustion; I—excess air; I and I—main and auxiliary air heater

other purposes, say, for the heating and desalination of water, heating of fuel oil, drying of solid fuel, etc. Such heat utilization also solves some problems in the boiler plant proper, for instance, it decreases the temperature of waste gases, raises the boiler efficiency, or increases the efficiency of electrostatic precipitators.

The heating of surplus air is utilized advantageously in multi-purpose plants where the heat-transfer medium—air is free from impurities and

can be heated to a high temperature. In such plants, the heat-transfer medium (combustion products) has a substantially lower temperature and the gases are much more pure and have a lower corrosion activity. Surplus hot air can be produced in various air heater circuits (Fig. 19.23). Tubular air heaters are preferable, since they can deliver clean, ash-free air and are less prone to clogging by ash at low temperatures of waste gases.



HEAT EXCHANGE IN HEATING SURFACES OF BOILERS

20.1. Thermal Characteristics of Water Walls

Heat absorption by water walls in a boiler furnace is determined mainly by the radiant heat transfer from high-temperature gases in the furnace to the external surface of water wall tubes which may be covered by a layer of ash. In open-type furnaces with an uprising flame, the heat absorption by water walls due to convection can be neglected, since gas velocities at furnace walls are low, while deposits on the tube surface offer a large thermal resistance to heat transfer. In furnaces with a turbulent flame (cyclone primary furnaces, furnaces with intercepting jets, etc.), the convective component is significant and should be taken into account.

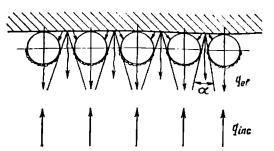
The intensity of heat exchange by radiation is expressed by the Stefan-Boltzmann law which relates the heat flux density to the area of heat-absorbing surface and the fourth power of temperature. The incident heat flux

from the flame core onto the water walls in boiler furnaces, q_{inc} , is usually equal to 400-700 kW/m², which forms a temperature gradient $\Delta t =$ $= t_{ex} - t_{in} = 150-350 \text{ deg C across a}$ relatively thin layer of surface deposits on the tubes, i.e. the temperature on the surface of deposits, t_{ex} , substantially exceeds the temperature of the tube wall metal t_{ω} . Thus, the intensity of radiant heat exchange between high-temperature gases and the surface of water walls in a boiler furnace depends neither on temperature nor on the pressure of the working fluid in the boiler.

The zone near a water wall is under the combined influence of the incident, effective and absorbed (resulting) heat fluxes. The incident heat flux from the flame core, according to Stefan-Boltzmann's law, can be written in the following form:

$$q_{inc} = a_{fi}c_0 (T_{fi}/100)^4 \cdot 10^{-3} \text{ kW/m}^2$$
(20.1)

Fig. 20.1. Formation of effective heat flux on a water wall



The effective heat flux (Fig. 20.1) is the sum of the intrinsic radiation from deposits on the tube surface and from the refractory lining, which have a sufficiently high temperature, and the reflected flux, since the coefficient thermal radiation from the surface of the walls and the deposits is less than unity:

$$q_{ef} = q_{intr} + q_{reft}$$

$$= [a_{dep}c_0 T_{dep}^4 \cdot 10^{-11}]$$

$$(1 - a_{dep}) q_{inc}] x$$

$$+ [a_{ref}c_0 T_{ref}^4 \cdot 10^{-11}]$$

$$+ (1 - \sqrt{a_{ref}}) q_{inc}] (1 - x) (20.2)$$

In formulae (20.1) and (20.2): T_{fl} , T_{dep} , T_{ref} are the temperatures of the flame, the external layer of deposits and the refractory lining, K, a_{fl} , a_{dep} , and a_{ref} are the emissivities of these elements, c_0 is the emissivity of the black body, $W/(m^2K^4)$, and x is a coefficient determining the fraction of the total radiation which falls on the surface of the water walls; this is essentially the ratio of the radiationabsorbing surface of a water wall to the surface of the furnace occupied by that wall.

The effective radiation from heatabsorbing surfaces in boiler furnaces is rather high and may reach 50-60% of the incident heat flux.

The difference between the incident heat flux and the effective heat flux is what is called the radiant heat flux absorbed by a heating surface, q_r, which is further transferred to the working fluid:

$$q_r = q_{inc} - q_{et} \tag{20.3}$$

$$\psi_{ef} = q_e/q_{inc} \qquad (20.4)$$

characterizes the fraction of heat absorbed by heating surfaces on furnace walls. It is called the coefficient of thermal efficiency of a water wall. With a higher value of ψ_{ef} , water walls operate more efficiently, i.e. they absorb a higher fraction of the total heat. According to the results of tests in boiler furnaces, ψ_{ef} has rather stable values in combustion of similar fuels and constitutes: $\psi_{ef} = 0.4-0.45$ for solid fuels, 0.5-0.55 for fuel oils, and 0.65 for natural gas. It should be noted that the thermal efficiency of water walls in a boiler furnace is not constant along their height: it is higher at the level of the flame core and diminishes towards the exit from the furnace. For refractory-faced water walls from studded tubes, $\psi_{ef} = 0.2$ -0.25; for bottom water walls covered by a layer of fireclay brick, $\psi_{\sigma f} =$ = 0.1. The patterns of variation of the incident, effective and absorbed heat fluxes along the height of a furnace are shown in Fig. 20.2.

The fraction of heat flux falling on a water wall is also determined by the angular coefficient x of a water wall. As may be seen from Fig. 20.1, the angular coefficient x cannot be found merely from geometric considerations as the ratio of the projection of the area of all tubes onto the refractory lining of the wall to the total area of that wall. Part of the incident heat absorbed by the wall lining is then re-radiated onto the rear surfaces of tubes and is thus utilized. Only a small fraction of the heat flux from

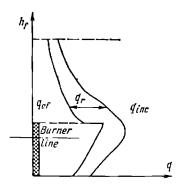


Fig. 20.2. Variations of the incident and effective heat flux along the furnace height

Refractory-faced portion of water walls is shown hatched

the lining within the angle of vision of the flame ($<\alpha$) is reflected back into the furnace volume. With a denser spacing of the tubes (smaller tube pitch $\sigma_t = s/d$), the angle α is narrower, and therefore, a higher fraction of the total heat flux falls onto the tubes of a water wall (Fig. 20.3). With a common density of tube spacing $\sigma_t = 1.05-1.1$, the angular coefficient is equal to 0.97-0.99. Theore-

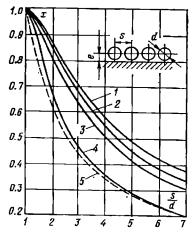


Fig. 20.3. Angular coefficient of a singlerow smooth-tube water wall

1—for a water wall on furnace side, including the radiation from refractory facing, at $e/d \ge 1.4$; 2—ditto, at e/d = 0.8; 3—ditto, at e/d = 0.5; 4—ditto, at e/d = 0.5; 5—for a tube row in a membrane or platen (radiation of refractory facing neglected)

tically, with densely arranged tubes $(\sigma_t = 1)$ or with a refractory-faced water wall, x = 1, i.e. the heat flux falls entirely onto a water wall.

The ratio

$$\xi = \frac{T_{fl}^4 - T_{dep}^4}{T_{fl}^4} = 1 - \left(\frac{T_{dep}^4}{T_{fl}^4}\right) \quad (20.5)$$

is called the coefficient of relative fouling of water wall tubes. Since $T_{dep} >$ > 0, the coefficient ξ is always less than unity and is lower at a higher T_{dep} , i.e. at a higher thickness or thermal resistance of deposits. The difference $T_{fl}^* - T_{dep}^*$ determines the absorbed heat flux, and therefore, the coefficient ξ characterizes the fraction of the incident radiant heat which is absorbed by working fluid in the tubes. Thus, the coefficient of thermal efficiency ψ_{ef} of a water wall and the coefficient of fouling ξ are correlated as follows:

$$\psi_{ef} = x\xi \tag{20.6}$$

The coefficient ξ is somewhat higher than ψ_{ef} , since it disregards the small fraction of heat radiated onto the lining of furnace walls.

In boiler furnace calculations, the concept of a radiant heat absorbing surface is often used:

$$H_r = x F_w^{ww} \tag{20.7}$$

where F_w^{ww} is the surface area of the furnace that is occupied by water walls, \mathbf{m}^2 .

The radiant heat absorbing surface is thought of as a continuous plane surface which has the same temperature values, degree of fouling and emissivity as the tubes of the water wall. As follows from (20.7), H_r is somewhat smaller than the surface area of the walls on which heating tubes are arranged, but greater than the sum of projections of these tubes onto the wall.

The ratio of H_r to the total surface area of furnace walls

$$\chi = H_r/F_w \qquad (20.8)$$

is called the coefficient of furnace screening.

In most cases, all the walls of a boiler furnace are covered by heating tubes, except for small portions around the burner ports, manholes and viewholes, and the coefficient of furnace screening is usually equal to 0.95-0.96 and approaches the magnitude of the angular coefficient x. In low-capacity boilers, some portions of the furnace walls are left uncovered by tubes and the coefficient χ is substantially lower.

20.2. Flame Emissivity

As regards their intensity of radiation in the visible spectrum, flames may be luminous, semi-luminous or nonluminous. This division is rather conditional, since radiation is a flow of radiant energy not only in the visible region. The radiation of a luminous or semi-luminous flame is determined by the presence of solid particles (coke, soot and ash) in the flow of combustion products. Radiation of a non-luminous flame is mainly due to triatomic gases (CO₂ and H₂O) present in the furnace. Their radiation is selective and falls mainly on the region of thermal (infra-red) wavelengths. At the same temperature, gaseous substances have a much lower radiation intensity than solids. The radiation in the volume of a boiler furnace is essentially the sum of the radiations of solid particles and gases and depends on the kind of fuel being burned.

The intensity of radiation of solid particles in a flame depends on particle size, particle concentration in the furnace volume, and the properties of the solid.

Coke particles are usually of a size $\delta_c=10\text{-}250~\mu\text{m}$. Their radiation intensity is close to that of black body radiation, but their concentration in the flame is not high (less than 0.1 kg/m³). They are concentrated mainly near burners, so that their radiation onto furnace water walls is only 25-30% of the total radiation in the furnace.

Ash particles are mostly of the same size as coke particles, but are distributed all over the furnace volume. Their concentration in the gaseous medium depends on the ash content of the fuel being used. The intensity of their radiation constitutes 40-60% of the total radiation in the furnace. The intensity of radiation is lower at a higher temperature of the gaseous medium and increases as the gases are cooled.

Soot particles can form in large quantities on combustion of fuel oils and natural gas. They have a high concentration in the flame core and possess a high emissivity [10].

Radiation of triatomic gases in the furnace is determined by their concentration and the thickness of the radiating volume. The emissivity of a gaseous medium is determined by the relationship following from Bouguer's law:

$$a_g = 1 - e^{-h} g^p p^s$$
 (20.9)

where k_g is the coefficient of absorption in the gaseous medium, p_p is the total partial pressure of triatomic gases, MPa, and s is the effective thickness of the radiating layer, m, which may be found from formulae given in [20].

The emissivity of the gaseous medium in boiler furnaces $a_g=0.4$ -0.5 and their radiation is roughly 20-30% of the total radiation in the furnace. For all solid fuels, the flame emissivity in the boiler furnace is found by the formula:

$$a_{fl} = 1 - e^{-k_f ps}$$
 (20.10)

where k_f is the effective coefficient of absorption in the furnace and p is the pressure of gases in the furnace, MPa.

In pulverized-coal combustion, the lum inous flame fills almost the whole volume of the furnace (Fig. 20.4). The process of burning extends along almost the whole height of the furnace, while the presence of incandescent ash particles makes the flame brightness almost the same along the whole height. The coefficient of absorption

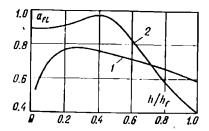


Fig. 20.4. Variation of flame emissivity along the height of furnace 1—for pulverized coal; 2—for fuel oil

of radiation by the furnace volume is calculated by the formula:

$$k_{l} = k_{g}r + k_{a}\mu_{a} + k_{c}\varkappa$$
 (20.11)

where $r = r_{\rm RO_a} \div r_{\rm H,O}$ is the total volume concentration of triatomic gases, k_a is the effective coefficient of absorption by ash particles, μ_a is the dimensionless concentration of ash in furnace gases, k_c is the effective coefficient of absorption of radiation by coke particles, and \varkappa is a constant for a particular kind of fuel (low-reactive or high-reactive).

Solid fuels produce a semi-luminous flame on combustion.

In fuel-oil fired boilers, the radiation of the flame varies greatly along the furnace height (Fig. 20.4). Intensive radiation by soot particles is concentrated in the zone of the flame core and quickly decreases beyond that zone. For this reason, in calculating the emissivity of a flame a_{fl} , the flame is conditionally divided into two portions: luminous and non-luminous:

$$a_{fl} = ma_{lum} + (1 - m) a_g$$
 (20.12)

where a_{lum} is the emissivity of the luminous portion which can be found from formula (20.10) upon substituting k_{lum} for k_l , a_g is the emissivity of the non-luminous gaseous medium which is found by formula (20.9) and m is an averaging coefficient which is determined as the fraction of the furnace volume that is occupied by the flame core.

The averaging coefficient for natural gas combustion is m = 0.1 and

for fuel oil, $m \Rightarrow 0.55$. With solid fuels, since the flame is extended along the whole furnace and is equally bright in all its portions, it is taken that m = 1.

The radiation of the luminous portion of the flame is mainly due to burning soot particles and additionally to high-temperature triatomic gases. The effective absorption of radiation in the luminous portion of the flame is:

$$k_{lum} = k_s + k_e r \qquad (20.13)$$

where k_4 is the absorption by soot particles.

The intensity of radiation from the flame core on combustion of fuel oil is 2-3 times that for solid fuels and, even assuming that the averaging coefficient m = 0.55, the heat absorption by water walls in a fuel-oil fired furnace is higher, as noted earlier (see Sec. 8.1). Thus, as a boiler furnace is changed from pulverized coal to fuel oil, the temperature of gases at the furnace outlet decreases noticeably. In natural gas combustion, heat absorption by the water walls is mainly determined by the radiation of nonluminous triatomic gases, which is less intensive, so that the heat absorption by less fouled water walls is close to that in the combustion of solid fuel.

The coefficient of thermal radiation in a furnace, a_f , is determined by the emissivity of the flame a_{fl} and the thermal efficiency ψ_{ef} of the water walls:

$$a_f = \frac{1}{1 + \left(\frac{1}{a_{ff}} - 1\right) \psi_{ef}}$$
 (20.14)

An increase in a_{fl} leads to a higher absorption of radiant heat in the furnace. An increase in the thermal efficiency of the water walls, ψ_{ef} , implies an increase in their heat absorption, and therefore, a decrease in the effective radiation into the furnace volume, the result being a lower thermal radiation in the furnace and lower heat flux falling onto the water walls.

In that case the mean heat flux absorbed by a water wall

$$q_r = c_0 a_f \psi_{ef} \left(\frac{T_{fl}}{100}\right)^4 \cdot 10^{-3} \text{ kW/m}^2$$
(20.15)

increases somewhat due to a relatively greater change in the coefficient ψ_{ef} . In formula (20.15), T_{fl} is the mean effective temperature of the gaseous medium in the furnace, K.

20.3. Calculation of Radiant Heat Transfer in a Furnace

Heat transfer from the flame to the heating surfaces arranged on the walls of a furnace is highly complex. The heat transfer process occurs here in parallel with fuel combustion which forms internal heat sources in the radiating medium. The gas temperature distribution along the furnace height is determined by the ratio between the intensities of heat release and heat absorption. Additionally, thermal characteristics of water wall tubes may vary owing to surface fou-

At the initial stage just after ignition, intensive burning of fuel ensures a rise in temperature of the gases. At the same time, the flow of heat energy to the water walls increases. At a certain distance from the burners, the temperature attains a maximum corresponding to the equality between the heat release and heat absorption. Further, heat release decreases rapidly and becomes less than the level of heat absorption, so that the temperature of the gases diminishes monotonically. The rate of its decrease depends on the temperature maximum in the flame core, the presence or absence of fuel afterburning in the top portion of the furnace, and the degree of fouling of the water walls.

The method for analysis of heat exchange in boiler furnaces developed in the USSR is based on the combined use of the results of analytical and experimental studies [20]. It proceeds from the possibility of application of the theory of similarity which fairly well describes the principal thermal characteristics of boiler furnace operation and their connection to furnace design.

The calculation is based on the semiempirical formula (A. M. Gurvich) which can be written in the following dimensionless form:

$$\theta_{j}^{*} = \frac{B_{0}^{0.6}}{B_{0}^{0.6} + Ma_{j}^{0.6}} \qquad (20.16)$$

It relates the dimensionless temperature of gases at the furnace outlet. 07, with the Boltzmann number Bo which characterizes the ratio of the heat released on fuel combustion to the maximum intensity of heat removal to the surface of the water walls. The coefficient M in the formula is introduced to consider the pattern of the temperature field in the furnace volume. The dimensionless temperature of gases is found from the formula:

$$\theta_f' = T_f'/T_a \tag{20.17}$$

i.e., it is essentially the ratio of the gas temperature at the furnace outlet, T_I^* , K, to the adiabatic temperature T_a , K. It is always less than unity and shows how the temperature of gases in the furnace volume decreases due to heat exchange.

The Boltzmann number in formula (20.16) can be found by the following formulae depending on the original data available for the calculation:

$$Bo = \frac{\varphi B_w V_g c_g}{c_0 \psi_{ef} F_{to} T^2} \qquad (20.18a)$$

$$Bo = \frac{\varphi B_w V_g c_g}{c_0 \psi_{ef} F_w T_a^3}$$
 (20.18a)

$$Bo = \frac{B_w Q_r}{c_0 \psi_{ef} F_w T_a^4} \frac{1}{1 - \theta_f^2}$$
 (20.18b)

 $(B_w$ is the rated fuel consumption). Let us consider in more detail the heat calculation and the role of various terms in formulae (20.16) through (20.18).

The principal thermal characteristics of a furnace are the useful heat release Q, and the enthalpy of gases at the furnace outlet, I_t . The useful heat release is the sum of the available heat of burned fuel Q_a^w (minus the heat losses q_3 , q_4 and q_6 within the furnace), the heat introduced into the furnace by preheated and cold (sucked in) air $(Q_a = Q_{ha} + Q_{ca})$, and the heat of the portion of flue gases recirculated from the convective shaft back into the furnace, Q_{rec} :

$$Q_{f} = Q_{a}^{w} \frac{100 - q_{3} - q_{4} - q_{6}}{100 - q_{4}} + (Q_{a} - Q_{a,cx}) + Q_{rec}$$
(20.19)

The quantity of heat Q_{ha} is calculated by the temperature of hot air at the outlet from the air heater. If air is proheated before the air heater by an external heat source (say, by bleeder steam from the turbine), Q_a should be diminished by the quantity of that heat $Q_{a,cx}$ (see Sec. 7.2).

The enthalpy of gases at the outlet from the furnace, I_I^{σ} , can be found on the I, ϑ -diagram or from tables for a given gas temperature ϑ_I^{σ} (see Sec. 8.1).

If the useful heat release of the furnace, Q_j , could be transferred completely to the products of combustion, i.e. without heat exchange with the heating surfaces (adiabatic conditions), we would obtain the highest (theoretical) temperature of combustion that could be developed in the furnace; it is more often termed the adiabatic temperature of combustion:

$$\vartheta_a = \frac{Q_f}{(Vc)_{av}} \tag{20.20}$$

where $(Vc)_{av} = Q_f/\vartheta_a$ is the average specific heat of combustion products formed by 1 kg of fuel within the temperature interval $0-\vartheta_a$, kJ/(kg K). As may be seen, to find the adiabatic temperature, we have first to pre-estimate it. The adiabatic temperature of gases depends on the kind of fuel (in particular on its calorific value) and excess air ratio and is equal to 1700-1850°C for brown coals and peat and 1850-2100°C for coals, anthracite, fuel oils and natural gas.

The characteristic temperatures of a boiler furnace are the adiabatic temperature and the temperature of gases at the furnace outlet. They serve as reference points in the calculation of heat exchange in furnaces. The calcu-

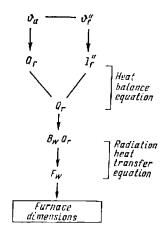


Fig. 20.5. Sequence of heat calculation of boiler furnace

lation of a boiler furnace is done in the following sequence (Fig. 20.5). Upon determining Q_f and I_f , the heat transferred by radiation in the furnace, Q_r , kJ/kg, is found. After that we can calculate the surface area of the furnace covered by water walls, F_w , to absorb the total quantity of heat B_wQ_r at the given temperature conditions (\mathfrak{F}_a and \mathfrak{F}_f) and determine the thermal efficiency of the water walls. Finally, we determine the dimensions of the furnace to arrange the water walls.

Thus, the calculation of heat exchange in a boiler furnace is based on solving two principal equations:

the heat balance equation

$$Q_r = \varphi (Q_f - I_f^r) = \varphi (Vc)_g \times \times (\vartheta_a - \vartheta_f^r)$$
 (20.21)

and the equation of radiant heat exchange

$$B_w Q_r = c_0 a_j x F_w (T_l^4 - T_{dep}^4) \cdot 10^{-11}$$
(20.22)

where T_{dep} is the temperature on the surface of the deposits on the waterwall tubes, K, φ is the coefficient of heat retention (see Sec. 6.2), and $(Vc)_g$ is the averaged heat capacity of gases in the temperature interval $(\vartheta_a - \vartheta_l^*)$, kJ/(kg K). Equation (20.22)

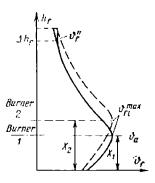


Fig. 20.6. Temperature field in gases along the height of furnace

Burner 1. Burner 2—burner lines in the furnace;

X₁. X₁—heights of burner lines

can be transformed as follows:

$$B_w Q_r = c_0 a_f x F_{to} T_{fl}^4 \left(1 - \frac{T_{dep}^4}{T_{fl}^4} \right) \cdot 10^{-11}$$
(20.23)

The expression in the brackets is the coefficient of fouling ξ ; noting further that $x\xi = \psi_{ef}$, we finally get: $B_wQ_r = c_0a_f\psi_{ef}F_wT_M^{\epsilon}\cdot 10^{-11}$ (20.24) from which the surface area of the water walls is found as

$$F_w = \frac{10^{11} B_w Q_r}{c_0 a_f \psi_{ef} T_{11}^4} \qquad (20.25)$$

The accuracy of the calculation of a boiler furnace depends heavily on how accurately the effective temperature of gases in the furnace, T_{fl} , has been averaged. The actual temperature field of gases has a rather intricate pattern along the furnace height (Fig. 20.6). This variation in the flame temperature is described satisfactorily by the formula of three parameters:

$$T = T_{\mathbf{o}} \sqrt{\Lambda X} t e^{-pX} \tag{20.26}$$

which covers the entire diversity of temperature fields possible in boiler furnaces. In this formula, A, t and p are the parameters of the temperature field in the furnace, T and T_a are the current and adiabatic temperatures of gases, K, X is the relative height of the position of the flame zone with the highest temperature of burning in the design height of the furnace. The

solution of equation (20.26) is quite complicated and can be found in the specialist literature [10].

In view of the difficulties associated with the mathematical description of temperature fields in furnaces, attempts have been made to find semiempirical solutions for T_{fl} by using the two characteristic temperatures T_f^a and T_a . The most successful is the solution found by G. L. Polyak and S. N. Shorin, which is given as a relationship of the dimensionless relative temperatures of gases:

$$\theta_{II} = \sqrt[L]{m} (\theta_{II})^n \qquad (20.27)$$

where $\theta_{fl} = T_{fl}/T_a$ is the dimensionless average effective temperature of gases, and m and n are empirical coefficients which depend on the conditions of combustion and gas cooling in a furnace.

Upon substituting the dimensionless ges temperatures, equation (20.27) can be written in the following form:

$$T_{tt}^4 = mT_a^{4(1-n)}(T_{tt}^n)^{4n}$$
 (20.28)

Numerous experimental studies of heat transfer in boiler furnaces have demonstrated that the coefficient m, which characterizes the similarity of temperature fields in boiler furnaces under different conditions of combustion, is only slightly dependent on the operating conditions of a furnace, but is closely associated with the coefficient n. Coefficient m is close to unity and constant for a given value of n. The exponent n is a function of the relative position of the temperature maximum of gases in a furnace (Fig. 20.7). The most typical values of X

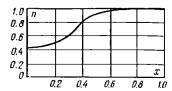


Fig. 20.7. Variations of the exponent n as a function of the position of the maximum of gas temperature X in the furnace

for boiler furnaces (X = 0.15-0.30) fall on the region where n depends substantially on X. In this connection, the methodics of heat transfer calculation for boiler furnaces is revised periodically (every few years) when new types of boilers appear.

For boiler furnaces of small dimensions and for combustion chambers in two-chamber and open-type furnaces of high-capacity boilers, it may be taken that n=0.5. Then the average temperature of gases can be determined by the formula:

$$T_{II} = 0.925 \sqrt{T_a T_I^2}$$
 (20.29)

For open-type single-chamber furnaces of steam boilers of power stations, it is taken that n=2/3 and m=1 which corresponds to X=0.33. With a lower position of the burners in the furnace, and consequently, of the zone of the highest temperature of gases (Fig. 20.6), the temperature of gases at the outlet of a furnace of a given design will be lower, so that in the calculation by the assumed value of T_I^* , the height of the furnace should be diminished.

Since the actual position of the highest temperature of flame may deviate from the design value, an additional temperature field parameter is introduced into formula (20.25):

$$M = A - BX$$

where A and B are empirical coefficients depending on the kind of fuel used.

In most cases, the relative position of the highest temperature zone in the furnace, X, coincides with the level of burners X_b , since with horizontally directed burners the maximum of temperature in the flame core lies in the horizontal portion of the flame. In all cases of delayed ignition or combustion (straight-flow burners, multitier burners), a correction ΔX is added to X_b to account for a higher actual position of the flame core; then $X = X_b + \Delta X$.

The final formula for determining the surface area of a boiler furnace is as follows:

$$F_{w} = \frac{\frac{10^{11} B_{w} Q_{r}}{c_{0} a_{f} \psi_{cf} M T_{f}^{2} T_{a}^{3}}}{\frac{1}{M^{2}} \left(\frac{T_{a}}{T_{f}^{2}} - 1\right)^{2}} (20.30)$$

Check calculations of heat transfer in boiler furnaces of given dimensions can also be made by using the dimensionless formula (20.16) which can be transformed as follows:

$$\frac{1}{1+M\left(\frac{a_f}{Bo}\right)^{0.61}} - 273$$
 (20.31)

The fraction of heat absorbed by the heating surfaces in the furnace due to radiation, relative to the useful heat release in the furnace, is called the coefficient of direct heat release:

$$\mu_I = Q_r/Q_I \tag{20.32}$$

Disclosing Q_r from formula (20.21), we obtain:

$$\mu_f = \varphi_t^* \left(1 - \frac{I_f^*}{Q_f} \right) \qquad (20.33)$$

The direct heat release to the heating surfaces in the furnace is usually roughly half of Q_f . It is higher in the combustion of dry solid fuels, natural gas and fuel oils which have a high temperature of burning ($\mu_f = 0.45$ -0.55) and lower for moist brown coals ($\mu_f = 0.38$ -0.45); for fuels with a very high moisture content (W > 7), the direct heat release may be less than 0.35. In all cases, μ_f decreases with an increase of θ_f , as follows from formula (20.33).

Zonal calculation of furnace. The general heat-transfer calculation enables us to find only the average heat absorption by furnace walls. In many cases, however, especially for once-through boilers, it is essential to know the heat absorption in individual zones of the furnace so as to determine the temperature conditions in tubes (say, in the lower, medium and upper radiant sections). This can be done by zonal calculation of a furnace: the furnace is divided into a number of

zones (four-six), each 3-6 m in height. The first zone is that of the highest heat release (the zone of burners).

In zonal calculation, the temperatures of the gases at the outlet from each zone are determined, after which the heat fluxes in the zones can be calculated. First, the heat balance is written for the zone of the highest heat release so as to find the temperature of gases at the outlet from that zone:

$$Q_{1}^{"} = B_{w} (Vc)^{"}\theta_{1}^{"} = B_{w} (\beta_{bu}Q_{1}^{u} + Q_{ph} + Q_{a} + Q_{rec} - Q_{6}) - B_{w}Q_{r1}$$
(20.34)

where $\beta_{bu} = 0.92$ -0.98 is the degree of fuel burn-up in the zone and Q_{11} is the radiant heat of the gases that is absorbed by the water walls and by the upper and lower horizontal planes which confine that zone.

The term $B_{w}Q_{r1}$ is calculated from formula (20.22) by assuming an approximate value of T_{f11} which will then be determined more accurately. Upon solving equation (20.34), one can determine the temperature of gases at the outlet from the first zone, ϑ_{1}^{x} .

After that, gas temperatures at the outlet of the second and subsequent zones are found by considering the additional heat of fuel burn-up in a zone, $\Delta \beta_{bu} Q_l^w$ and the radiant heat given up to the water walls. Having made the calculation for all the zones, one can

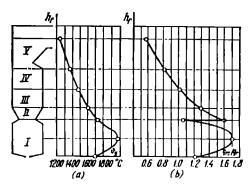


Fig. 20.8. Curves of zonal calculation of furnace

(a) variation of gas temperature along the furnatee height; (b) variations of relative heat fluxes construct the curve of gas temperature variation along the furnace height (Fig. 20.8a). For the given temperature conditions in the zones, one finds the absorbed radiant heat fluxes in the furnace zones, q_{rz} [by formula (20.15)]. The ratio of q_{rz} for a particular zone to the absorbed heat flux of the whole furnace, q_r , (Fig. 20.8b) is called the coefficient of distribution of heat absorption along the furnace height:

 $\eta_h = q_{rz}/q_r \qquad (20.35)$

20.4. Radiant Heat Transfer in Boiler Flue Ducts

The heating surfaces of a boiler just downstream of the furnace are swept by high-temperature gases and receive an appreciable portion of heat by radiant heat transfer. In particular, semi-radiant heating surfaces of a boiler, such as the platen superheater or slag screen, which are arranged at the outlet from the furnace, receive the major portion of their heat by absorbing direct radiation of the flame core.

The radiant heat absorbed by platens from the furnace, $Q_{r,p,l}$, is found as the difference between the radiant heat flux at the inlet to a platen and the heat flux re-radiated from the platen onto subsequent heating surfaces:

$$Q_{r,pl} = Q_{r,ln} - Q_{r,oul}$$
 (20.36)

The radiant heat of the furnace that is absorbed in the inlet plane of platens is found by the results of heat calculation of the furnace:

$$Q_{r,in} = \beta \eta_h q_r \frac{H'_{r,pl}}{B_r}$$
 (20.37)

where β is the coefficient taking account of the mutual heat exchange between the furnace medium and the gases in the volumes between platens; it is taken equal to 0.6-1.0 depending on the kind of fuel and the gas temperature at the furnace outlet; and $H'_{r,pl}$ is the surface area of the inlet section of platens that absorbs radiant heat from the furnace, \mathbf{m}^2 .

The heat flux at the outlet from platens, $Q_{r,out}$, kJ/kg, is the sum of the radiant heat that has passed from the furnace through the platens (re-radiated heat) and the heat that is radiated from the gas volume in the platens onto the subsequent heating surfaces:

$$Q_{r.out} = Q_{r.in} (1-a) \frac{\varphi_{pl}}{\beta} + Q_{r.pl}^*$$
(20.38)

where a is the emissivity of gases in the zone of platens, φ_{pl} is the angular coefficient of radiation from the inlet onto the outlet section of the platens, which is determined by their geometrical characteristics (the spacing between the platens and the depth of platens), and $Q_{r,pl}^{r}$ is the radiation of gases from the platen zone onto a heating surface behind the platens, which can be found from the laws of radiant heat exchange.

The heat absorption by the platens from the gas flow is determined by the coefficient of heat transfer α_1 , $kW/(m^2 K)$, which considers both the radiant and convective heat transfer in a platen:

$$\alpha_1 = \xi (\alpha_r + \alpha_c) \qquad (20.39)$$

where the coefficient ξ is introduced to account for the non-uniform sweeping of the platens by gases.

Direct radiation from the furnace into the zone of platens increases the temperature on the surface of the deposits on the front row of the platen tubes and decreases the heat absorption from the gas flow sweeping these tubes. This circumstance is considered in the calculation of the coefficient of heat transfer in platens, k_{pl} , by introducing a multiplier $(1 + Q_{r,pl}/Q_{pl})$ into the denominator, which gives the following formula for k_{pl} :

$$k_{pl} = \frac{1}{\frac{1}{\alpha_1} + (1 + Q_{r,pl}/Q_{pl}) \left(\epsilon + \frac{1}{\alpha_2}\right)}$$
(20.40)

where $Q_{p,l}$ is the total heat absorption by the platens due to convective heat transfer and to radiation of gases in the platen zone.

For other heating surfaces of a boiler, radiant heat transfer is determined only by radiation from the space between the tubes, i.e. they receive no direct radiation from the furnace. The emissivity of gases depends on their temperature, the intensity of thermal radiation in the gas volume, the size of that volume, and the temperature and emissivity of external deposits on the heating surfaces. The emissivity in a convective heating surface can be found from the formula:

$$\alpha_r = \frac{q_r}{T_g - T_{dep}} \tag{20.41}$$

where T_g is the average temperature of gases, K, and T_{dep} is the temperature of the deposits on heating tube surfaces, K.

The radiant heat flux absorbed by the unit surface area of a heating surface in a convective gas duct can be found from the equation:

$$q_r = c_0 a_g \frac{a_{dep} + 1}{2} (T_g^n - T_{dep}^n) \cdot 10^{-11}$$
(20.42)

where a_g and a_{dep} is the emissivity of the gaseous medium and external deposits on the tube surface (it is usually taken that $a_{dep} = 0.8$). The exponent n is equal to four in a dust-laden gas flow (to consider the radiation of ash particles) and n = 3.6 for clean gas flows (on combustion of natural gas or fuel oil). The temperature of the surface of the deposits on superheater tubes is determined by the formula:

$$T_{dep} = T_{wf} + (\epsilon + 1/\alpha_2) \frac{B_{wQ}}{H}$$
 (20.43)

where T_{wf} is the temperature of the working fluid (steam), K, and Q is the total heat absorption by the heating surface, including the radiation from intertubular space, kJ/kg.

As may be seen, the equation for T_{dep} includes another unknown quan-

tity, the intertubular radiation, so that the solution should be found by the successive approximation method. For other heating surfaces arranged in the zones of lower gas temperatures and lower heat fluxes, the difference between T_{dep} and T_{wf} decreases and T_{dep} can be found from the formula:

$$T_{dep} = T_{wf} + \Delta t \qquad (20.44)$$

where Δt is the recommended calculation norm for various types of heating surfaces and temperature zones.

The emissivity of a gaseous medium in the general case is expressed by Bouguer's relationship:

$$a_g = 1 - e^{-hps}$$
 (20.45)

where the product kps is called the total optical thickness of combustion products (see also Sec. 20.2).

The coefficient of absorption of radiant heat by a gaseous medium on combustion of solid fuels is determined (with the concentration of fly ash taken into account) by the formula:

$$k = (k_g r - k_a \mu_a)$$
 (20.46)

For ash-free gas flows, the second term is equal to zero. In convective tube banks, the thickness of the intertubular radiating layer s depends on the relative tube pitches s_1/d and s_2/d [20] and is usually equal to 0.1-0.2 m, i.e. it is only 1/20-1/50of the thickness of gas layer in the furnace. For this reason, radiant heat transfer in convective banks (considering also that the temperature of gases decreases in flue ducts) is two or three orders of magnitude lower than that in the furnace. At gas temperatures below 400°C, radiation in dense tube banks can be neglected.

Gas volumes in front of convective banks have a more noticeable radiation intensity owing to a higher effective thickness of the radiating layer. In this case, the emissivity for a convective bank arranged behind such a volume is higher than in the calculation of intertubular radiation and can he found from the formula:

$$\alpha_r' = \alpha_r \left[1 + A \left(\frac{T_g}{1000} \right)^{0.25} \left(\frac{l_p}{l_b} \right)^{0.07} \right]$$
(20.47)

where α_r is the heat transfer coefficient determined by the relationships of intertubular radiation, Λ is a coefficient depending on the kind of fuel, l_v and l_b are the depth of the gas volume (along the gas motion) and of the tube bank, m, and T_g is the gas temperature in the volume upstream of the bank, K.

The heat transferred by radiation from a gas volume onto a tube bank upstream of that volume is negligible, since the temperature of gases in the volume is lower than the average temperature in the bank. This radiation is not considered in our calculations.

20.5. Convective Heat Transfer in Boiler Flue Ducts

Convective heating surfaces of a boiler include the elements arranged in the horizontal gas duct behind the semi-radiant surfaces of platens or slag screen and those mounted in the convective shaft. This is a zone of relatively low gas temperatures where the effectiveness of radiant heat transfer decreases rapidly. In order to increase heat absorption by convection, gas velocities in this zone are increased and the heating surfaces are extended by using tube-coil banks with densely spaced tubes arranged for cross-current sweeping by the gas flow. The intensity of heat absorption by convective heating surfaces diminishes along the gas path from 40 kW/m^2 for the convective superheater to 1-2 kW/m² for the air heater. In regenerative air heaters, gases and air move along heat-transfer plates, so that the intensity of heat transfer per unit area of smooth surface of plates is only 1/3 or 1/4 that in tube-coil banks. Heat transfer can be intensified by various methods (see Ch. 19). The principal equations of heat transfer are as follows:

heat transfer equation:

$$Q_f = \frac{k\Delta t II}{B_r} \qquad (20.48)$$

heat balance equation for the gas sido:

$$Q_b^g = \varphi (I_g' - I_g'' + \Delta \alpha I_{su}^0) \quad (20.49)$$

and equation of heat absorption by the working fluid:

$$Q_b^{wf} = \frac{D}{B_r} (i'' - i')$$
 (20.50)

An additional condition is that $Q_I =$

 $= Q_b^g = Q_b^{uf}, \text{ kJ/kg.}$ In equations (20.48) through (20.50), H is the heating surface area of an element, m^2 , k is the heat-transfer coefficient, $kW/(m^2 K)$, Δt is the rated temperature gradient, K, B_r is the rated fuel consumption, kg/s, D is the flow rate of the working fluid, kg/s, I'_g and I''_g are the enthalpies of combustion products at the inlet to and outlet from an element, kJ/kg, I_{sa}^{o} is the enthalpy of sucked-in air, kJ/kg, Δα is the relative air inleakage in a gas duct, i' and i" are the enthalpies of the working fluid at the inlet to and outlet from the element. kJ/kg.

For the air heater, the heat absorption by working fluid (air) is determined by the following formula instead of (20.50):

$$Q_{8}^{\alpha} = (\beta_{\alpha} - 0.5\Delta\alpha_{\alpha h}) (I_{h\alpha}^{0} - I_{e\alpha}^{0})$$
(20.51)

where β_a is the relative excess of air at the inlet to the air heater, and I_{ha}^{0} and I_{ca}^{0} are the enthalpies of the theoretical volume of air taken respectively at the temperature of hot and cold air, kJ/kg.

In design calculations, the heating surface area H is found from equation (20.48). In this case, the heat absorption by working fluid through surface area H is known, so that one can determine from equation (20.49) the enthalpy and temperature of gases behind that surface and then the temperature gradient Δt and the heattransfer coefficient for the heating surface.

In a check calculation, one has to determine the heat absorption by the heating surface from equation (20.48). To calculate Δt and k, one has to assume the heat absorption Q_b approximately and then to correct it, sinco the specified value and that obtained from equation (20.48) should differ by not more than 2%.

In the heating surfaces of boilers, heat transfer occurs through a multilayer cylindrical wall—a metallic tube with deposits on its outside and inside surfaces. The heat-transfer coefficient is usually determined by using the formula for a plane multi-layer wall. With a sufficiently large tube diameter, this involves no substantial error and simplifies the calculation considerably.

The formula for the heat-transfer coefficient of a plane multi-layer wall is as follows:

$$k = \frac{1}{1/\alpha_1 + \delta_{sl}/\lambda_{sl} + \delta_{m}/\lambda_m + \delta_{sc}/\lambda_{sc} + 1/\alpha_2}$$
(20.52)

where α_1 and α_2 are the coefficients of heat transfer from the heating fluid to the wall and from the wall to the heated fluid, δ_m , δ_{sl} , and δ_{sc} are the thicknesses of the metallic wall, the external layer of deposits (slug) and the internal layer (scale) and λ_m , λ_{sl} , and \(\lambda_{ec}\) are the respective conductivi-

The ratios $1/\alpha_1$ and $1/\alpha_2$ are essentially the thermal resistances to heat transfer at the external and internal side of the surface. For heating surfaces in which tho working fluid is water, a steam-water mixture or superheated steam of supercritical pressure, the coefficient α_2 substantially exceeds (by two or three orders of magnitude) the coefficient α_1 , so that the thermal resistance on the internal side $1/\alpha_2$ can be neglected, since $1/\alpha_1$ is much greater than $1/\alpha_2$. The same is true of the thermal resistance of the metallic wall: δ_m/λ_m is much lower than $1/\alpha_1$.

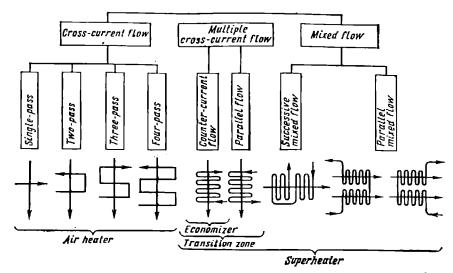


Fig. 20.9. Diagrams of motion of the heating and heated media in the convective elements of boilers

For the normal operation of a boiler, the scale on the internal surface of the tubes should cause no noticeable increase in thermal resistance, so that the thermal resistance of scale is not considered in calculations $(\delta_{sc}/\lambda_{sc} = 0)$.

The thermal resistance of deposits on the outside surface of the tubes $\varepsilon = \delta_{al}/\lambda_{al}$ (ash, slag, soot, corrosion products) cannot be disregarded, though measures are taken in boiler operation to remove deposits periodically (see Sec. 16.1).

In certain cases when some of the quantities in the relationships indicated are unknown or the flow conditions are too complicated, the coefficient of effectiveness \(\mathbb{Y} \) of a heating surface is used, which can be found from the heat balance of that surface.

The heat-transfer coefficient is determined as an average value for the whole surface considered at an average velocity of gases. The coefficient of utilization ξ is also introduced to take account of the incomplete or non-uniform sweeping of a heating surface by the gas flow. In view of the above, formula (20.52) can be re-written in the following form:

for tube-coil surfaces

$$k = \frac{1}{1/\alpha_1 + \varepsilon + 1/\alpha_2}$$
 (20.53a)

and for regenerative air heaters

$$k = \frac{\xi}{1/\alpha_1 + 1/\alpha_2}$$
 (20.53b)

Diagrams of the relative motion of heating and heated fluids in most typical heating surfaces of boiler plants are shown in Fig. 20.9. The temperature gradient, i.e. the average temperature difference between the heating and heated fluid for the whole heating surface in parallel flow, counter-current flow and multiple cross-flow (with the number of passes more than four) can be found by the formula:

$$\Delta t = \frac{\Delta t_g - \Delta t_l}{2.3 \log \Delta t_g / \Delta t_l} \quad (20.54a)$$

where Δt_g is the greatest and Δt_l is the lowest temperature difference between the fluids at the ends of a heating surface, deg C.

If the ratio $\Delta t_g/\Delta t_l$ does not exceed 1.7, the temperature gradient can be determined sufficiently accurately as the mean arithmetic value.

As shown in Fig. 20.9, a multiple cross-current flow of fluids is used pre-

dominantly in the heating surfaces of boilers. With the number of passes for one of the fluids less than four or with a mixed mode of flow, heat transfer in a heating surface will be somewhat lower than in a purely counter-current motion of fluids; this can be considered by introducing a correction coefficient Ψ for the temperature gradient determined for the counter-current mode:

$$\Delta t = \Psi \Delta t_{cc} \qquad (20.54b)$$

The coefficient Ψ is determined graphically for a particular scheme of motion of the two fluids [20].

In circuits with parallel or successive-mixed fluid flow, the calculation can be carried out separately for each of the passes and determines the average temperature gradient for the whole heating surface:

$$\Delta t_{av} = \frac{\Delta t_1 H_1 + \Delta t_2 H_2}{H_1 + H_2} \quad (20.55)$$

where H_1 , H_2 are the areas of individual portions, m^2 , and Δt_1 , Δt_2 are the temperature gradients in them, deg C.

The same approach is used when the heat capacity of one of the fluids changes significantly within the heating surface calculated, for instance, if the fluid changes the state of aggregation (as in a boiling economizer). In this case, the heat absorptions Q_1 and Q_2 in the portions with the single-phase and two-phase fluid are found separately, after which the temperature gradients Δt_1 and Δt_2 are determined for these portions and averaged for the whole heating surface.

20.6. Velocities of Gases and Working Fluid in Convective Heating Surfaces

The optimal velocity of gases is that at which the cost of construction and operation of a heating surface is at a minimum. With an increase in gas velocity, the coefficient of heat transfer increases, with the result that the heating surface is smaller and its cost is lower. A smaller heating surface offers a lower hydraulic resistance to the motion of working fluid (steam, water, steamwater mixture) in tubes, which makes it possible to reduce the power of the feed pumps (in the design of superheaters and economizers) and increase the power of the turbine (due to a lower resistance of the reheater). On the other hand, this increases the consumption of energy needed to overcome the greater resistance of the gas path.

The optimal velocity of gases can be found by technico-economical calculation. The best version is that in which the expected expenditures Z are at a minimum:

$$Z = C + p_r K$$
 (20.56)

where C is the annual expenditures associated with operation of the equipment, p_r is the coefficient of depreciation for the capital expenditures (in power engineering, it is usually taken equal to 0.15 per year), and K is the initial capital expenditure:

$$K = K_{el} + K_{dr} + K_{sub}$$
 (20.57)

where K_{el} is the cost of the considered element in the gas path, K_{dr} is the cost of the draft fan, and K_{sub} is the cost of a substituting auxiliary plant to provide continuous power generation in the system.

Gas velocities calculated by this method are called economical (w_{ec}) . For instance, for balanced-draft steam boilers with the cost of reference fuel of 10-12 (roubles/t), the economical gas velocities, m/s, in superheaters and economizers with staggered tubes are as follows (at the rated boiler load):

For in-line tubes, the economic gas velocities are 40% higher.

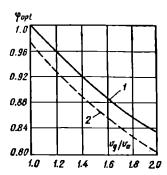


Fig. 20.10. Curves of optimal ratio of air and gas velocities in air heaters

1—at specific heat ratio $c_a/c_g = 1$; s—at $c_a/c_g = 0.9$

In solid fuel combustion, these velocities in most cases are not reached in view of excessive abrasion wear of tubes by ash (see Ch. 16).

The intensity of heat transfer and the heating surface area in air heaters depend on the velocity of gases and air, since the heat-transfer coefficients at both sides (gas and air) are rather close to each other, especially in regenerative-type air heaters. For this reason, one first has to determine the optimal velocity ratio $\varphi_{opt} = w_a/w_g$. Figure 20.10 shows variations of φ_{opt} which depend on the ratio of volumes and specific heats of gas and air in regenerative air heaters.

As has been demonstrated by calculations, φ_{opt} depends on the type of air heater. For tubular air heaters, $\varphi_{opt} = 0.4\text{-}0.55$ at the optimal gas velocity $w_{ec} = 11 \pm 2$ m/s; for regenerative air heaters, $\varphi_{opt} = 0.86\text{-}0.92$ at $w_{ec} = 10 \pm 1$ m/s [53].

The velocity of working fluid (steam, water) has no essential effect on the intensity of heat transfer and on the dimensions of the heating surfaces, since the highest thermal resistance appears on the side of heating gases.

The velocity of steam in superheaters should be such as to ensure reliable conditions for the operation of the metal, which in the final result determines the cost of a heating surface. All superheaters can be divided into three main types which require different approaches to the selection of velocities [6].

1. The last (outlet) banks of the superheater and reheater are arranged in the high temperature zone of gases and have the highest temperature of steam. The high velocity of steam ensures the reliable cooling of tube walls; at any load of the boiler, the metal temperature is not allowed to rise above the limit for a particular steel. Pressure loss is of minor importance in this case. This group also includes radiant and semi-radiant (platen) superheaters which receive high heat fluxes by direct radiation from the furnace with a high degree of nonuniformity across the furnace width.

2. Convective superheaters are arranged in zones with gas temperatures below 1 000°C and cooled by moderately superheated steam. These have a certain reliability margin of the metal. In this case, the optimal steam velocity is determined mainly by the superheater design and by the minimal cost of a tube bank (for a lower steam velocity, one has to increase the number of tubes in a bank or the diameter of the tubes, which will increase the dimensions, and therefore, the cost of a superheater section).

3. Reheater banks are arranged in zones of moderate gas temperatures (450-600°C). Their optimization is carried out by considering the effect of steam velocity on the dimensions of a heating surface and on the hydraulic resistance of the reheater (a higher hydraulic resistance decreases the power of the turbine).

Practical recommendations can be reduced to specifying the mass velocities of steam $w\rho$, $kg/(m^2 s)$, for various types of heating surfaces. For instance, the recommended values are $w\rho = 800$ -1 100 kg/(m² s) for the outlet convective banks and 450-600 kg/(m² s) for other banks. For reheater banks, $w\rho = 250$ -400 kg/(m² s) is sufficient.

The velocity of water in convective economizers is selected so as to prevent stratification of the steam-water flow should boiling occur and to ensure carry-over of gas bubbles along the tubes, which appear in heated water. On the other hand, it should be considered that an increase in water velocity can substantially raise the hydraulic resistance and the energy con-

sumption by feed water pumps. The recommended values of water velocity are $w\rho = 500\text{-}600 \text{ kg/(m}^2\text{s})$; in the economizer zone where partial evaporation of water can take place (the boiling section of the economizer), $w\rho = 800\text{-}900 \text{ kg/(m}^2\text{ s})$.



LAYOUT AND HEAT CALCULATION OF STEAM BOILER

21.1. Boiler Layout and Structures

Boiler layout. The mutual arrangement of the gas ducts of a boiler and the direction of combustion products in them determine what is called the boiler layout which may be Π -, T-, N-, U-shaped, four-pass or tower-type (Fig. 21.1).

Among them, the II-shaped layout is most popular (Fig. 21.1a): the furnace is arranged in an ascending shaft and the convective heating surfaces are arranged in a parallel descending shaft. Its advantage is that fuel is supplied and gases are removed at the bottom, which is convenient for the removal of liquid slag and for the shot-blasting of the convective surfaces. The draft fans are mounted on the zero (floor) level so that their vibrations are not transferred to the boiler structure. A drawback of the II-shaped layout is that, since the gases are turned by 180°, the heating of convective surfaces and the concentration of ash over the cross section of the convective shaft may be uneven.

The T-shaped layout is often employed in high-capacity boilers, with two convective shafts arranged at both sides of the furnace (Fig. 21.1b); this

makes it possible to diminish the depth of the convective shafts and the height of the connecting horizontal ducts. The total cross-sectional area of the two convective shafts is higher than in the former scheme. The draft fans are mounted on the zero level. The T-shaped layout is especially suitable for the boilers fired on fuels which produce abrasive ash (such as Ekibastuz coals) where it is essential to reduce the velocity of combustion products. The removal of combustion products from the two shafts in this layout involves. however, certain design difficulties. Furthermore, a T-shaped boiler is more intricate in design than a II-shaped one and consumes more metal.

In some cases, the three-pass layout, with draft fans arranged at the top, is employed (Fig. 21.1c); this scheme is popular, in particular, in the FRG. In this layout, combustion products move ascendingly in the furnace and convective shaft and descendingly in the connecting vertical gas duct. The four-pass layout (Fig. 21.1d) is used in boilers firing fuels with a very high ash content (such as oil shales). These schemes are characterized by the provision of an intermediate vertical gas duct which may have heating surfaces (say, platens) with widely spaced tubes

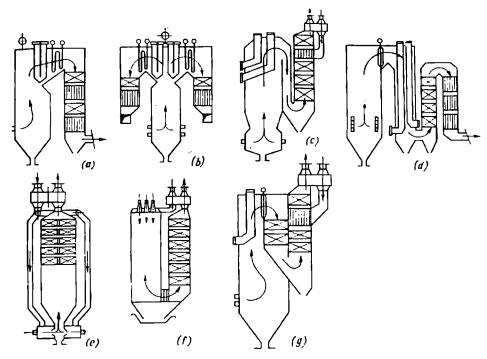


Fig. 21.1. Various schemes of boiler layout

so as to prevent slagging in the high-temperature zone.

High-capacity supercharged boiler plants are often built by the tower scheme (Fig. 21.1e) in which combustion products move in the furnace and convective shaft only upwards. This layout offers the following advantages: the boiler plant occupies the least area in plan: convective heating surfaces are swept evenly by the combustion products, since there are no turns of gas flow; the gas path has the lowest resistance, since there are neither descending gas ducts nor gas turns. The drawbacks of this layout are as follows: the boiler structure should be stronger to carry the convective heating surfaces; the last banks of the superheater and the draft and exhaust fans must be installed at a high height; it is impossible to employ shot-blasting for cleaning the convective heating surfaces. There is a compromise solution: the semi-tower layout in which the regenerative air heater and draft means are mounted on the floor level and connected with the boiler tower by a descending gas duct which has no heating surfaces.

The tower layout requires a heavier boiler structure to carry the air heaters, draft fans and stack. It is more suitable for supercharged fuel-oil fired boilers and for boilers operating on high-ash brown coals: since ash-laden combustion products move in a straight path, abrasion wear of convective surfaces is not as strong as in other types of boiler layout.

In the U-shaped two-pass layout (Fig. 21.1f), combustion products move downwards in the furnace and upwards in the convective shaft (inverted layout). Burners are arranged in the furnace roof. The U-shaped layout has the following advantages: the flame adequately fills in the furnace volume; the superheater can be mounted on a low level, i.e. it can be connected

with the turbine by a shorter pipeline; and the air path has a low aerodynamic resistance (the air heater is quite close to the burners). Its drawbacks are as follows: fuel must be supplied to the burners to an appreciable height; draft and exhaust fans and ash collectors must be mounted on a high level. The U-shaped (inverted) layout can be employed for the combustion of gas, fuel oil and solid fuels provided that the furnace is of the dry-bottom type.

The layout of high-capacity boilers of monobloc units should meet certain additional requirements (in view of the considerable size of the plant in plan, the application of steam reheating, etc.) and onsure reliable operation of the boiler in combination with the turbine.

The span of roof beams in highcapacity boilers is sometimes diminished by dividing the furnace and convective shafts into two sections, i.e. a boiler has two housings (each with its own metal structure and refractory setting) in which the heating surfaces are arranged symmetrically (two-housing symmetrical boiler layout). Provision of shut-off valves on the two symmetrical sections gives what is called the double-bloc layout. With one of the sections shut down, a double-bloc boiler plant can operate at 50-% load, which improves its manoeuvrability, but increases the cost and the unit fuel consumption, since the superheater has the same rated hydraulic resistance. Further, the turbine efficiency decreases at partial loads.

In two- and multi-pass schemes of gas motion, the furnace and gas ducts may either be fully separated from each other (as in Fig. 21.1a, b, d, and f) or have a common longitudinal partition formed by gas-tight water walls (Fig. 21.1c and g).

Air heaters are arranged either in the lower portion of the convective shaft or outside the boiler or even the boiler room, so as to leave space for the arrangement of burners or shotblasting installation. The arrangements of the particular heating surfaces of boilers have been described in more detail in Chs. 17-19.

Boiler structure. Modern high-capacity boilers manufactured in the USSR are usually of the II- or T-shaped layout. The layout of a boiler determines the design of the boiler structure and stress distribution in it.

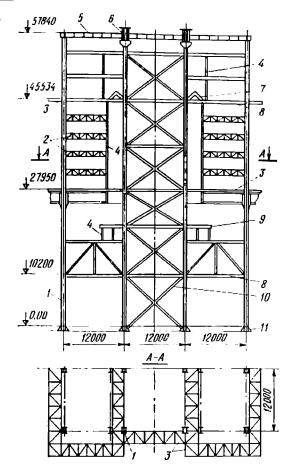
The boiler structure is essentially a supporting steel framework which carries various boiler elements, such as the drum, heating surfaces and headers, refractory setting, insulation and casing, pipelines and ducts, stages, stairs, etc. The boiler structure may be either isolated from the load-carrying structure of the boiler room building and be supported on an individual foundation or made integral with those structures. In the former case, the boiler structure carries the whole load of the boiler and transmits it to tho foundation. The load on the foundation includes the mass of the boiler proper and its structure and the mass of the working fluid: water and steam. In southern regions where an open or semi-open layout of boiler equipment is allowed, the boiler structure should also withstand the wind load, and often, seismic loads.

Figure 21.2 shows schematically a once-through boiler (type Π -57, D== 1650 t/h) of the T-shaped layout mounted on a separate foundation. The boiler structure consists of vertical columns, horizontal beams and support trusses, main beams, roof ceiling, ties, and stands. All elements of the structure are joined by welding.

The number of supporting columns depends on the steam-generating capacity of a boiler. Columns are usually mounted only at the corners of the furnace and convective shaft. Additional columns may be installed in highcapacity boilers having an appreciable width or depth. All columns are connected at the top by transverse beams or trusses. They increase the boiler structure stability, prevent the buckling of columns, and serve to support the drum (in drum-type boi-

Fig. 21.2. Structure of a oncethrough boiler type P-57 (D = 1 650 t/h) with an independent foundation

1-column; [2-vertical truss; 3-horizontal truss; 4-stand; 5-roof; 6-main beam; 7-inclined ceiling plate; 8-beam; 9-furnace frame; 10-ties; 11-store



lers), heating surfaces, and the supporting structures of stages. Additional stands and beams of smaller cross section are mounted to increase the structure rigidity and to fasten water walls, headers, hot-air ducts, etc.

The columns of a boiler structure transmit substantial concentrated loads onto the foundation. The unit pressure on the foundation is decreased by providing supporting shoes at the lower ends of the columns. Metal consumption for a boiler structure depends on boiler capacity and may constitute 0.8-1.2 kg per kg of hourly steam-generating capacity. Supporting columns and beams are not heated by gases; this insures more favourable operating conditions of the metal and

prevents substantial thermal stresses. Boiler structures are mostly made of carbon steels, such as Grade St. 3. Some less critical elements may be made of steel Grade St. 0.

The introduction of gas-tight all-welded water walls (see Sec. 17.3) with lighter refractory setting has made it possible to substantially reduce the boiler mass (sometimes by 30-50%). A lighter boiler can thus be supported by the structures of the boiler room building. The drum and all heating surfaces of the furnace and convective shaft are suspended from heavy beams of the boiler room building roof and can freely expand downward. In this case, the boiler structure proper serves only to ensure the required ri-

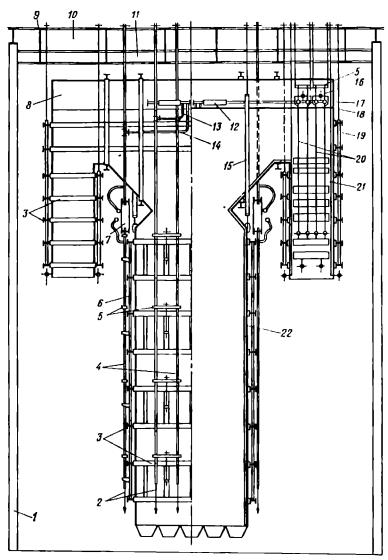


Fig. 21.3. Structure of gas-tight boiler type P-67 ($D=2\,650\,$ t/h) combined with boiler room

1—boiler room column; 2—working fluid to 1st pass of the lower radiation section; 3—reinforcing beams; 4—suspended tubes ('hot' suspensions); 5—cross-piece; 6—stand; 7—truss for suspending furnace side walls; 8—'hot lox'; 9—main beams; 10—intermediato beam; 11—lower beam; 18—working fluid mixer; 13—working fluid to suspended tubes of front (rear) furnace wall; 14—ditto, of a side wall; 13—slag screen; 16—discharge header of convective superheater; 17—discharge header of economizer; 18—suspended from intermediate beams; 19—suspended tubes of superheater and economizer (coincident); 21—water walls of convective furnace shaft; 22—furnace water walls

gidity and to fasten the boiler to the room building structures. In such designs, the boiler structure ensures the appropriate rigidity of welded water walls, absorbs the pressure in the gas path (in supercharged boilers), and carries stages and stairs. An example of the boiler structure carried by the room building is illustrated in Fig. 21.3. The boiler elements are suspen-

ded only from the main beams of the building and are not connected with the building walls, so that deformations of the building are not transferred to the boiler. The boiler elements are suspended from the main beams by means of numerous suspension ties. The heating surfaces are connected with the boiler structure their temperature expansions should be essentially of the same magnitude to avoid temperature stresses. This is ensured by means of 'hot' suspensions which are made from tubes supplied with the working fluid at a temperature roughly equal to that in the corresponding heating surfaces. In supercharged boilers, the pressure in the gas path is absorbed by horizontal beams spaced 2.5-3.0 m apart on all walls of the furnace, convective shaft and connecting gas ducts. These beams ensure proper rigidity of the boiler walls.

Suspended structures make it pos-

sible to dispense with separate boiler foundations, allow more space for mounting the auxiliary equipment, save much metal for the construction of the boiler structure, and reduce the time of construction work.

A boiler and its elements have definite dimensions in the cold state and occupy certain positions in space. In the working 'hot' condition, their dimensions and positions in space vary substantially. Thermal displacements are determined by the working temperature of the metal and the length of an element from its fixed end. With the coefficient of thermal expansion of the metal equal to 0.012×10^{-3} m/K, certain boiler elements may expand by 300 mm or even more.

Figure 21.4 shows the diagram of thermal expansions of elements in a suspended boiler type TGMP-204. All level marks relate to the cold state. Displacements have been calculated for temperatures of the metal in ope-

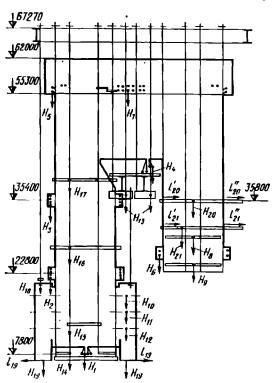


Fig. 21.4. Diagram of temperature expansions (H, mm; l, mm) of elements of a boiler type TGMP-204 (D = 2 650 t/h)

neaders of the bottom water wan, $H_1=290$; headers of water walls in parting joints, $H_1=208$; $H_1=130$; hower headers of the water wall in inorizontal gas duct, $H_4=70$; upper headers of water walls, $H_1=33$; lower headers of water walls in the convective furnace shaft, $H_4=200$; headers of convective superheaters, $H_4=29$; economizer headers, $H_4=145$, $H_4=187$; burners, $H_{10}=230$, $H_{11}=245$, $H_{12}=260$; vertical tubes of suspended tube system in horizontal gas duct, $H_{11}=138$; vertical tubes of furnace water walls, $H_{14}=248$, $H_{14}=224$, $H_{15}=103$; supporting plates on 'hol' suspensions, $H_{10}=247$; common wind box of furnace, $H_{10}=322$, $I_{11}=62$; headers of control surface of reheater, $H_{10}=119$, $I_{20}=8$, $I_{20}=105$, $I_{11}=142$, $I_{21}=4$, $I_{21}=57$

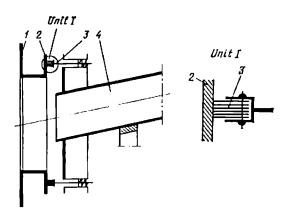


Fig. 21.5. Arrangement of burner in a port

J-furnace wall; 2-port flange; 3scaling sheets; 4-burner

ration. Vertical displacements are marked by H's and horizontal, by l's. In the example illustrated, horizontal displacements are marked only for the common box of the furnace, l_{10} , and for the headers of the control stage of reheater, l_{20} and l_{21} .

To avoid excessive thermal stresses, the structure of a boiler should have proper means to allow its free expansion on heating. A steam boiler is connected with its auxiliary equipment mounted externally of the boiler structure (pulverizing plant, feed water pipelines, etc.). It is also connected by pipelines with the turbine. Pipelines usually allow free expansion of the boiler owing to their self-compensation. Exceptions are pulverizing plants with mill fans, which have no dust pipelines of appreciable length, but are connected directly to the burners. In such cases, burner ports of a boiler are designed so as to move freely relative to the fixed burners and the gaps between them are sealed (Fig. 21.5).

'Hot box'. In high-capacity boilers, a great number of tubes and suspensions pass through the furnace roof to the outside, so that tight sealing of all of them is a problem. In such cases, the roof is covered by a tight steel shell which receives hot air from the air heater; this forms what is called the 'hot box'. Air pressure in a hot box is somewhat higher than the pressure of the combustion products at

the furnace top, which prevents the leakage of combustion products to the outside, but increases somewhat air inleakage.

The number of 'passes' from the hot box to the outside is much smaller than through the boiler roof; they mainly include the external pipelines and suspensions of headers. These 'passes' usually have silphon sealings.

When a hoiler is shut down for repairs, the time for natural cooling of the hot box is as much as 20-30 hours. Cooling can be accelerated by cold air blowing of the hot box.

Boiler enclosures. Boiler enclosures are important elements of boiler plants and consume much of work and materials for the manufacture. They have an essential effect on the conditions of operation of the furnace and convective shaft. In high-capacity boilers, the enclosure mass is so large that it greatly influences the design of the boiler structure and foundation.

In boilers of untight design, the enclosure is formed by continuous walls of ceramic materials, which isolate the gas path of the boiler from the surroundings. It is subjected to the action of hot combustion products which may carry ash, molten slag and unburned fuel particles. The enclosure is also subjected to pressure variations in the furnace (on flame break-off and fuel ignition), variable temperature stresses appearing on start-up and shut-down of the boiler.

stresses from temperature displacement of other boiler elements, static loads from various structures above it, and so on. The most dangerous are temperature stresses which appear on start-up and shut-down owing to uneven heating of boiler elements, so that various compressive, tensile and shear stresses may appear in the enclosure.

All these factors can hardly be considered properly in the design and construction of the enclosure. For this reason, reliable operation of the enclosure is ensured by selecting the appropriate materials which can operate under the heavy conditions indicated, and by certain design measures which can minimize the effect of these factors (composite enclosure, expansion joints, expansion compensators, etc.). The modern light enclosure of boilers has a mass of around 850 kg/m3. The mass of enclosure related to kg of hourly steam-generating capacity of a boiler is usually 0.4-0.5 kg or more. The enclosure of a highcapacity boiler consumes on the total more than 2000 t of ceramic materials. The boiler enclosure should have a high refractoriness, high mechanical strength, proper tightness, possess hewithstand at-insulating properties, temperature stresses and resist the action of ash and molten slag. The enclosure design is heavily dependent on the tube system of the heating surfaces in the furnace and convective shaft.

In modern power-station boilers, two types of enclosure are employed. The enclosure of the first type is attached immediately to tubes (on-tube enclosure), i.e. to tubular water walls of the furnace and gas ducts. The second type is made in the form of blocks or plates which are fastened to the boiler structure. Enclosure plates or blocks are manufactured industrially and mounted simultaneously with the boiler. The boiler enclosure consists of several layers: the first layer facing the gases and operating at temperatures of 1 500-1 800°C is made of a re-

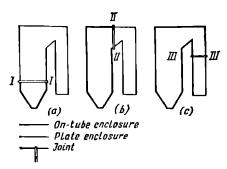


Fig. 21.6. Types of enclosure in various portions of a boiler

fractory material. Two or three layers beneath it, which operate at temperatures of 500-900°C, are made of heatinsulating materials. The outside surface of the enclosure is coated with gas-impermeable plaster (the allowable temperature for plaster coating is 100-200°C) or metal sheets. In gastight boilers with all-welded membrane water walls (see Sec. 17.3), light heat insulation over the tubes is quite sufficient.

In some boilers, the enclosure is of a combined type: plate enclosure in the prismatic portion of the furnace and on-tube enclosure in the dry-bottom hopper or inclined bottom. Enclosure blocks are suspended from roof beams of the boiler structure and can expand on heating together with the water-wall tubes. The two portions of the enclosure form a temperature joint around the furnace periphery, which should be properly sealed to prevent air inleakage (the horizontal plane I-I in Fig. 21.6a). The enclosure of convective gas ducts is usually of the plate type. If the furnace has an on-tube enclosure, a vertical joint is provided at the entry from the furnace to the horizontal convective duct (II-II in Fig. 21.6b). In a number of designs, ontube enclosure extends further into the convective duct and the temperature joint is provided in a horizontal plane of the convective shaft (III-III in Fig. 21.6c). If the on-tube enclosure extends further into the convective portion of a boiler, the temperature joint between it and plate enclosure operates at a lower temperature and may be simpler in design and more reliable.

Heat insulation. The boiler drum, headers, superheated steam, feed water and blow-down water pipelines, gas and air ducts, etc. are arranged beyond the boiler enclosure around the boiler, above the roof or along the boiler walls. These elements have a temperature of 200-600°C and are heatinsulated to protect the personnel against burns and minimize heat losses to the surroundings. By sanitary norms, the temperature on the external surface of heat-insulated elements must not exceed 55°C. A properly made heat insulation can diminish heat losses by 95-97% compared with a non-insulated surface. In addition, heat insulation of drums, headers, pipelines and fittings improves the temperature conditions for the metal, since it decreases the temperature gradient across the wall, and therefore, reduces temperature stresses. The external surface of heat insulation is made smooth and mechanically strong by applying cotton cloth and metallic casing or painting. Painting is also useful for marking various pipelines.

21.2. Thermal Diagram of a Boiler

All heating surfaces of a boiler communicate with one another in the gas and steam-water path, thus forming a complex system. Let us recall that the temperature of combustion products in certain points of the gas path and the temperature of water and steam in other points of the watersteam path should be maintained within definite limits to attain the highefficiency and reliability. For instance, the temperature of combustion products at the exit from the furnace is determined by the kind and characteristics of the fuol used (see Sec. 7.1). The temperature of waste gases is determined by technico-economical calculations (see Sec. 6.2). The temperature in the reheater zone sho-

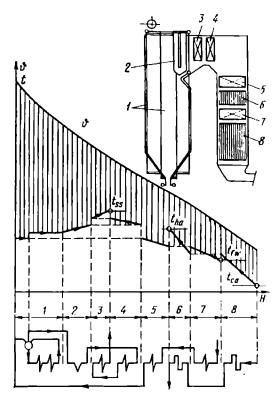
uld not exceed 850°C to avoid metal failure should steam supply to the reheater be occasionally stopped (see Sec. 18.2). The reliability of the last banks of the superheater, which operate under the most severe temperature conditions, is determined at the specified steam parameters by the allowable temperature of the superheater metal, which defines the design of the superheater and its position in a particular temperature zone along the gas flow (see Sec. 18.3). To ensure stability of hydrodynamic processes in intensively heated elements of oncethrough boilers, these elements must not be fed with a two-phase mixture. For this reason, the convective economizer should supply them with water of a certain degree of subcooling, rather than with a steam-water mixture, i.e. it must be of the non-boiling type (see Sec. 11.2). At supercritical pressures, reliable operation of heating surfaces in the zones of high heat capacity of the working fluid imposes specific requirements for their positioning in the furnace. For subcriticalpressure boilers, the transition zone, in which scale deposition is mainly concentrated, should be arranged in the zones of moderate hoating. The hot air temperature, which is determined by fuel characteristics, is essential for the selection of a particular type of air heater (see Sec. 19.3).

Thus, there is a system of limiting conditions, or reference points, for the temperatures of particular heating surfaces. Selection of these points means essentially the distribution of enthalpy increments between these heating surfaces and optimal positioning of these surfaces in the flow of combustion products, i.e. it means selection of the thermal diagram of a boiler.

The thermal diagram of a boiler is the schemo of arrangement of the heating surface banks along the gas flow and of arrangement of their pipelines. Two conditions are important for this selection: the working fluid at a higher temperature should be swept by the combustion products also

Fig. 21.7. Thermal diagram of a drum-type pulverized coal-fired boiler

I—evaporating heating surfaces (furnace water walls); 2—platen superheater; 3 and 4—hot and cold stage of convective superheater; 5 and 7—2nd and ist stage of economizer; 6 and 8— 2nd and 1st stage of tubular air heater



at a higher temperature, so as to maintain a high temperature gradient; it is advisable to employ counter-current flow of the working fluid and combustion products where possible. Theso conditions, however, are not always properly satisfied. For instance, the heating intensity of radiant heating surfaces arranged in the burner zone may be enormous, resulting in the low reliability of metal operation. For this reason, the heating surfaces to be arranged in such zones have a low temperature of the working fluid (preheating and evaporating surfaces and 'cold' superheater banks), while the last superheater banks are mounted in the zones of moderate temperatures of combustion products.

With a high heat absorption of particular heating surfaces (most often of superheaters), to enhance their reliability the heating surfaces are divided into a number of successively con-

nected sections, which diminishes the maldistribution of heat and ensures better intermixing of the flow (see Figs. 18.10 and 18.11).

In high-capacity boilers, the heat absorption by the water walls in the furnace is insufficient, which has led to the appearance of platens (see Sec. 7.1) and curtain walls (see Sec. 18.1). They diminish the temperature of combustion products at the furnace exit to the required level.

The superheated steam path in very powerful drum-type boilers and the entire water-steam path in once-through boilers are made in the form of a number of individually controlled flows. For convenience of control, the number of flows is usually two or four. Separation of the water-steam path into flows diminishes the non-uniformity of heat distribution across the boiler width and permits of using tubes of smaller diameter, but makes the

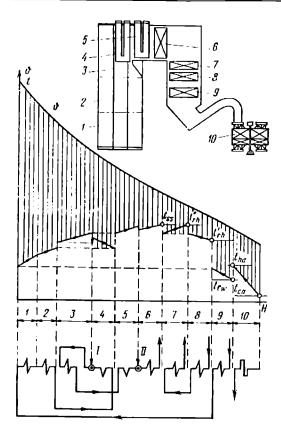


Fig. 21.8. Thermal diagram of a oncethrough gas and fuel oil fired boiler I—lower radiation section; 2—medium radiation section; 3—upper radiation section; 4—1st platen superheater; 5—2nd platen superheater; 5—convective superheater; 7 and 8—hot and cold reheater stages; 9—economizer; 10—regenerative air heater

boiler plant design more complicated and expensive, increases the number of fittings, and complicates automatic control.

Let us consider as examples the thermal diagrams of a drum-type and once-through boiler. The drum-type boiler (Fig. 21.7) has two stages of air preheating and accordingly two stages of heating the feed water which is fed to the drum from the second economizer stage. The evaporating water walls are arranged on the furnace walls and together with unheated downtake tubes form circulation circuits. Saturated steam separated in the drum is fed into the superheater which consists of a radiant section and platens connected in series along the steam path and of two convective banks connected by a mixed scheme, with the outlet bank arranged in the zone of a higher temperature. Water spraying to control the temperature of superheated steam is not shown in the diagram. The ordinates of the diagram (vertical hatching) give the temperature gradients in particular heating surfaces. As may be seen, they decrease significantly along the gas path.

In the once-through boiler whose thermal diagram is illustrated in Fig. 21.8, there is one stage of air preheating in an external regenerative air heater and one stage of feed water heating in the economizer. Preheated water is fed into the lower radiation section of the boiler from where it flows into the medium radiation section, the first bank of the platen superheater, the upper radiation section, the second bank of the platen superheater, and the convective superheater

which delivers the superheated steam of specified parameters. There are two water-spray means to control the temperature of live steam. Steam reheating is effected in two reheater banks connected by a mixed scheme and arranged in the convective shaft. Means for controlling the temperature of reheated steam are not shown in the diagram.

21.3. Heat Calculation of a Boiler

Aims and methods of heat calculation. One should distinguish between a design heat calculation and check calculation of a boiler, which differ from each other in their aims and the quantities to be determined.

Design calculation is undertaken in order to determine the dimensions of the furnace and other heating surfaces which can ensure the rated steam-generating capacity for the specified steam parameters and the desired efficiency and reliability of a boiler plant which uses a particular fuel and is fed with the foed water of a given temperature. The results of the design calculation are further used strength calculation and the selection of materials for particular boiler elements, hydraulic and aerodynamic calculations, and the selection of auxiliary equipment.

Check calculation is carried out for an existing or a design boiler. It is done for the given dimensions of the heating surfaces and the given characteristics of the fuel and detormines the temperatures of the working fluid. air and combustion products at the boundaries between the heating surfaces. Check calculation is also made when the temperature of feed water or superheated steam varies or a boiler changes over to another kind of fuel. An object of check calculation is to determine the thermal characteristics of a boiler and the possibilities for its control. In the design calculation, the dimensions of particular heating surfaces (say, of platens) can be chosen from the considerations of their

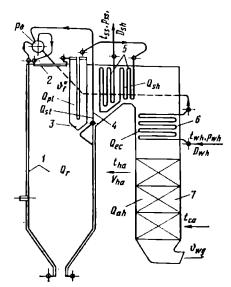


Fig. 21.9. Calculation diagram of a drumtype boiler

I—evaporating heating surfaces (furnace water walls); 2—roof superheater; J—platen superheater; 4—suspended tubes; 5 -convective superheater; 6—economizer; 7—air beater

arrangement in the boiler. In this case, a check calculation should then be made for these surfaces. Check calculation can estimate the efficiency and reliability of a boiler, provide data for recommendations on the boiler reconstruction and the initial data for hydraulic, aerodynamic and strength calculations.

The methodics of heat calculations of boilers are standardized [20].

Design heat calculation of a drumtype boiler. The procedure of the calculation will be demonstrated for the boiler shown in Fig. 21.9. First, we have to establish the theoretically required quantity of air and combustion products. The actual volume of air and combustion products in the furnace and gas ducts is calculated by considering the organized air supply to the furnace and air inleakages in the boiler of this particular design (with balanced draft). Then, the enthalpies of combustion products and air are determined. After that the heat balance of the boiler is calculated, from which we find the heat losses q_2 , q_3 , q_4 , q_5 , and q_6 , gross efficiency and fuel consumption.

The furnace is calculated as given in Sec. 20.3. Assuming the allowable heat release rate per unit area q_f , MW/m², we find the cross-sectional area of the furnace. The total surface area of furnace walls F_{to} is determined for the selected temperature at the furnace outlet θ_f . The last stage of the calculation is to check that the heat release rate per unit volume q_V is within the specified limits and that the assumed thermal efficiency ψ does not differ greatly from the calculated value (the allowable discrepancy is $\Delta \psi = \pm 0.05$).

Calculation of the heat exchange in the furnace should consider the heat absorbed by the platens (heating surfaces arranged at the furnace outlet) and by the radiant superheater section on the roof. Therefore, the dimensions of the platens and radiant superheater should be known before the calculation. Further, we determine the quantity of heat absorbed by the platens due to radiation from the furnace and to heat exchange within their zone, $Q_{\rm pl}$, and can then find the temperature of combustion products behind the platens, ϑ_{pl}^{ω} . Having considered the heat absorption by suspended tubes in the boiler, Qst, we can determine the temperature of combustion products before the convective superheater banks, O.h.

The remaining heat of the combustion products (upon their passage through the furnace, platens and radiant superheater) should be distributed between the convective heating surfaces of the steam-water path and the air heater. This heat is first distributed between the heating surfaces for which the inlet and outlet parameters of the working fluid are known or have been assumed. Thus we can find the quantity of heat that must be transferred to the superheater, Q_{sh} , to obtain the specified steam parameters $(D_{sh}, p_{ss}, \text{ and } t_{ss})$ and the quantity of heat to be given up in the air heater, Q_{ah} (to obtain V_{ha} and t_{ha}).

The rest of the heat should be allocated to the economizer for which the outlet parameters of water are not specified. Upon finding the quantities of heat transferred to the superheater and air heater, we can determine the enthalpies and temperatures of combustion products before and after the economizer.

The distribution of heat between the heating surfaces is then checked by the heat balance equation:

$$Q_{av}^{w}\eta_{gr} = (Q_r + Q_{pl} + Q_{sh} + Q_{gc}) \frac{100 - q_s}{100}$$
 (21.1)

The discrepancy in the balance should not be more than $\pm 0.5\%$ of the available heat Q_{av}^{ω} . If the chosen heat distribution between the heating surfaces is correct, we can go further to the design calculation of the superheater, economizer and air heater following the instructions given in Ch. 20.

Design calculation of a once-through boiler (see the diagram in Fig. 21.10). In once-through boilers, there are no definite zones of the particular states of aggregation of the working fluid. This circumstance should be properly considered when determining the zones of phase transition of the working fluid.

For the reliable operation of steamgenerating tubes in the boiler, the convective economizer should be of the non-boiling type (see Sec. 11.2), and therefore, a portion of the tube system in the furnace will operate as a radiant economizer (the inlet portion of the lower radiation section). In the boiler design considered (Fig. 21.10), the remaining portion of the lower radiation section functions as steam-generating tubes (at subcritical pressure). Steam superheating takes place in the top portion of the furnace (the upper radiation section), platens and convective superheater. The fixed values of the enthalpies in the reference points of the steam-water path allow us to determine the heat absorptions by the economizer surfaces and the lo-

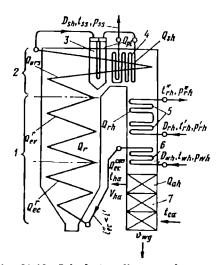


Fig. 21.10. Calculation diagram of a oncethrough hoiler

1—lower radiation section; 2—upper radiation section; 3—platen superheater; 4—convective superheater; 5—convective reheater; 6—economizer; 7—nir heater

wer and upper radiation sections, the remaining heat being distributed between the platens and convective heating surfaces.

As the working fluid moves along the water-steam path, its pressure decreases to a specified value at the outlet. The total pressure loss in the path is estimated as $\Delta p = (0.2\text{-}0.3)p_{ss}$, where p_{ss} is the superheated steam pressure, and can be corrected when needed by the hydraulic calculation.

The initial stage of the heat calculation, including the determination of heat losses and fuel consumption and the calculation of heat exchange in the furnace, is done following the same procedure as given earlier for the drumtype boiler.

The heat distribution between the heating surfaces is checked by the heat balance equation:

$$Q_{av}^{w}\eta_{gr} = (Q_{ee}^{e} + Q_{Irs} + Q_{urs} + Q_{pI} + Q_{sh})$$
(21.2)

The discrepancy should not exceed ±0.5% of the available heat. Further, the design heat calculation of the par-

ticular heating surfaces is carried out by the methods given in Ch. 20.

Check calculation of steam boiler. It is typical for the check calculation that there are no available data on the temperatures of waste gases (and therefore, on q_2). For this reason, one cannot determine the boiler efficiency and fuel consumption before the calculation. Without them, however, it is impossible to calculate further the furnace and convective heating surfaces. Thus, we have to assume the temperature of the waste gases the and that of the superheated steam t_{**} . Upon making the heat calculation. these temperatures must be checked and corrected, after which we can find η_{gr} and B. Thus, the check calculation is made by the method of successive approximations. With ngr and B being known and assuming a value of t_{ha} , we can begin the calculation of the furnace and all the heating surfaces. In the calculation of the furnace, we have to determine the actual temperature of combustion products at the furnace outlet, &. We first assume this temperature to perform the calculation and then correct it by formula (20.31). If the discrepancy is too high, the calculation is repeated for a new assumed value of The convective heating surfaces are calculated by the found temperatures (and enthalpies) of combustion products at the inlet to a particular surface and one of the known enthalpies of the working fluid (either at the inlet or at the outlet of that surface). Thus, among four sought-for quantities, two are usually known and the calculation of particular convective heating surfaces is done by the successive approximation method, presetting the value of Q_b for the gases or the working fluid and then comparing it with the calculated heat absorption Q_t (see Sec. 20.5). For the last heating surface in the gas path (air heater), the temperature of gases at the inlet is already known: $\theta_{ah} = \theta_{ec}$. We also know ϑ_{wg} (it has been preset earlier). With 0 found from the check calcu-

lation and with the known value of $Q_{ah},$ we can determine $artheta_{wg}$ and compare it with the assumed value. The calculation is considered to he completed if the two values of ϑ_{wg} differ by not more than ±10 deg C and the values of t_{ha} by not more than ± 40 deg C. Otherwise, the calculation is repeated for a new assumed value of ϑ_{wg} . A serious inconvenience of the successive approximation method is that the calculation for a particular heating surface must be done more than once to properly match the values of Q_t and Q_b . It can be substantially facilitated by using electronic computers.

Computer calculation of a boiler involves a laborious work for developing the mathematical means, model, algorithm and program of the heat calculation.

A developed mathematical model of

a steam boiler is essentially an interrelated system with three levels. The lower level includes computation units for particular heating surfaces (furnace, platens, convective surfaces, air heater). The medium level is a subsystem that determines the computation procedure for particular surfaces and their interrelations (along the gas and working fluid paths, injection lines, recirculation, etc.). The third (upper) level is a sub-system for the stabilization and optimization of the calculation.

An accurate check heat calculation of a boiler is only possible in large high-speed computers possessing a large capacity of the external and on-line memory; in that case the machine time of computation is around 10 minutes.



STEAM BOILERS OF HIGH-CAPACITY MONOBLOC UNITS

22.1. Selection of Boiler Design According to the Type, Capacity and Operating Conditions of Power Station

The principal trend in the development of thermal power stations is the application of monobloc (boiler-turbine) units whose capacity is steadily increasing. Monobloc units for a capacity of 300 MW with boilers producing 950-1 000 t steam per hour have been mastered. 500- and 800-MW units with single-housing boilers of a steam-generating capacity respectively of 1 650 t/h and 2 650 t/h have been constructed and put into service.

A recent achievement is a monobloc unit for a capacity of 1 200 MW with the boiler producing 3 950 t steam per hour.

As the unit capacity of steam boilers and especially of steam turbines increases, the unit capital cost (the cost of 1 kW of installed power) decreases noticeably; the staff coefficient (the number of personnel per unit power) decreases, too. This reduction however is substantial only for the power stations of the capacity up to 3-4 mln kW and becomes less essential for higher capacities (Fig. 22.1). On the other hand, as the steam-generating capacity of boilers in-

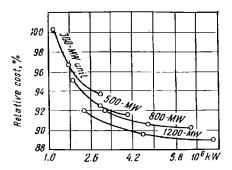


Fig. 22.1. Unit capital expenditures depending on power station capacity and unit power of plants

creases, the problem of reliability becomes more important. This should be quite clear, since boilers of higher capacity have more heating surfaces operating at supercritical or high pressures; accordingly, they have a greater total length of tubes and a larger number of welded joints which are probable points of emergency situations. Another probable cause of emergency situations is an elevated non-uniformity of heat distribution.

Maintaining the reliability of powerful monobloc units at the level already achieved in the existing smaller units presents a rather difficult problem. It places more rigorous requirements upon the quality of manufacture and the servicing of steam boilers.

An increase of the unit power of monobloc units is associated with an increase in the boiler and turbine dimensions. A pulverized-fuel fired boiler for an 800-MW unit may have a height up to 90 m or even more, which requires a corresponding increase in the height of the boiler room building. With a greater cross-sectional area of the boiler furnace and gas ducts, it is more difficult to properly organize the aerodynamics of gas flows. There are additional difficulties in the mounting and operation of high-capacity boiler equipment.

With the traditional design of the boiler heating surfaces and the convectional technology of their manufacture, there is a design barrier, as it were, which makes further growth of the boiler unit capacity problematic. A way out is to intensify heat exchange in the heating surfaces. The preliminary work has been carried out in modern metallurgy and boiler making to start the mass production of intensified profiles of heating surfaces which can change radically the design of particular boiler elements and the entire boiler plant. This can solve the problem of decreasing the boiler dimensions and increasing the reliability and efficiency of boilers. Intensified heating surfaces of boilers are finding ever wider application.

An increase of the unit capacity of monobloc units is usually associated with an increase of the capacity of power stations, and therefore, with a lower unit capital cost and accelerated rate of construction. As the capacity of thermal power stations increases, the effect of reduced unit capital cost becomes less pronounced (Fig. 22.1).

The efficiency of a monobloc unit and thermal power station depends substantially on the initial steam parameters. An increase in steam parameters involves, however, considerable technical difficulties. The possibility for increasing the temperature and pressure is determined, in the first place, by the availability of special steels which can operate reliably at higher parameters of superheated steam.

Pressure is one of the basic steam parameters and is decisive for the selection of the boiler type. At subcritical pressures, boilers of any type can principally be employed: oncethrough, drum-type with a gravity circulation or drum-type with a multiple forced circulation. The last type has found no wide use in the USSR. At thermal power stations for subcripressures, natural-circulation tical drum-type boilers are usually employed. The driving circulating head decreases with increasing pressure, so that the highest pressure in the drum is established at the level of 17 MPa so as to ensure reliable circulation. A higher pressure may cause a loss of reliability. This limit can be raised somewhat by using boiler tubes with internal helical fins (rifled tubes) which increase turbulization of the fluid at the tube walls.

The decisive effect of pressure on the selection of the type of boiler is limited by the monobloc capacity. With a very high capacity of monobloc units, the cost of the drum, which is the most expensive and metal-consuming element of a drum-type boiler, becomes too high. For instance, the mass of the drum for a 500-MW boiler plant may be as high as 200 t, which involves considerable difficulties in mounting and operating the boiler. For this reason, the capacity of monobloc units with drum-type boilers for supplying base loads does not usually exceed 400-500 MW. At supercritical pressures, the sole solution is the use of once-through boilers which are now built for the unit capacities up to 4 000 t/h (for 1 200-MW monobloc

The superheated steam temperature is even more important than pressure for the selection of the boiler design and especially that of the superheater (see Fig. 18.12).

For the same steam temperature, the metal temperature of the superheater tubes is higher than that in the pipeline and the difference depends on the heating intensity. The usual trend in the design of superheaters is to use various steel grades in the superheater proper and in the pipelines which connect the superheater sections and the boiler with the turbine, so as to minimize the cost of metal and ensure the required reliability.

When selecting the parameters of superheated steam, it should be taken into consideration that the cost of metal in the superheater and pipelines increases with an increase in temperature (Fig. 22.2). This increase in the cost may be different and depends on the temperature zone where the metal

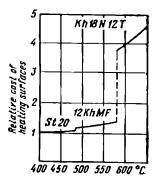


Fig. 22.2. Relative cost of heating surfaces as a function of nominal wall temperature (the cost of steel Grade 20 (0.25 C) at a wall temperature of 450°C is taken as unity) and temperature ranges of application of various steels

K b18N12T-18Cr-10N1-T; 12KbMF-0.12C-Cr-Mo-V; S120-0.25C

operates. For instance, a change from pearlitic low-carbon steel to low-alloyed steel of the same class in the temperature zone of 450-550°C involves no substantial increase in the cost of the superheater, since the low-alloyed steel has a higher strength, and therefore, superheater tubes may have a smaller wall thickness. Pearlitic steel can however operate reliably only at temperatures up to 580-585°C (the steam temperature 545°C). At higher temperatures, more alloyed austenitic steels must be employed which are much more expensive (5-8 times) and thus greatly increase the capital expenditures. On the other hand, within the temperature limits of the applicability of a particular steel grade, an increase in temperature causes only a slow rise of capital expenditures (due to a decreased allowable stress and increased metal consumption). For this reason, it is guite important to ensure a high coefficient of heat transfer at the steam side of the superheater, especially in its hot bank and radiant section, so as to lower the metal temperature. In view of the possibilities offered by pearlitic steels, new boilers are designed for a steam outlet temperature of 560°C at a subcritical pressure of 14 MPa (drum-type boilers) or 545°C at a supercritical pressure of 25.5 MPa (once-through boilers). Steam temperature behind the reheater is usually taken at the level of the temperature of live steam or slightly higher (545-565°C), since the pressure in the reheater is significantly lower.

The kind and quality of fuel have an essential effect on the boiler design. Fossil solid fuels are characterized by the presence of ash. The composition and temperature characteristics of ash determine the method of fuel combustion and slag removal. They also should be considered when selecting the temperature of combustion products at the furnace outlet. This temperature should be selected so as to form granulated ash particles in the convective gas ducts and thus avoid the slagging of the heating surfaces. With a higher temperature at the furnace outlet, water walls in the furnace may have a smaller area, and therefore, the furnace may be smaller. On the contrary, a lower temperature of the combustion products at the furnace outlet requires an extension of water walls and an increase in the dimensions of the furnace and heating surfaces of the boiler plant, since the convective surfaces will operate at reduced temperature gradients.

The velocity of combustion products in convective gas ducts must be limited to avoid quick wear of the heating surfaces by ash; this is associated with a reduced intensity of heat exchange, which necessitates an increase in the area of heating surfaces and the metal consumption. A decrease in the velocity of combustion products, especially in high-capacity plants with high flow rates of fuel and combustion products, requires an essential increase in the cross-sectional area of gas ducts, which may lead to a high non-uniformity of velocities and temperatures of gas flows and poor reliability of the boiler. This also makes the boiler design more complicated. A way out is to use a more intricate and heavy T-shaped boiler layout (see Fig. 21.1b).

In gas and fuel-oil fired boilers, the heating surfaces are not subject to wear by ash, which makes it possible to increase the velocity of combustion products and intensify the operation of the heating surfaces. Thus, fueloil fired hoilers are more compact than those operating on pulverized coal. A change to fuel oil or gas can raise the reliability and efficiency of a boiler plant. Since mineral impurities (ash) are present in fuel oils only in low concentrations and are not contained in natural gas, one can choose the optimal and economically most favourable (or close) temperature at the furnace outlet to minimize the cost of boiler; this temperature will be much higher than the temporature in pulverized-coal boilers to prevent the slagging of heating surfaces. For gas and fuel-oil fired boilers this temperature is usually taken to be 1 250°C. With such a high temperature of combustion products at the furnace outlet, the fraction of radiant heat exchange in the furnace is not high and, since the conditions of fuel combustion allow a higher heat release rate in the furnace volume, it is possible to decrease the total area of water walls, and therefore, the dimensions oſ the furnace, in particular, its height.

A high temperature of combustion products in the convective gas ducts, in combination with a higher allowable velocity (to avoid wear of heating surfaces), makes it possible to intensify heat transfer and heat absorption and thus to diminish the cross-sectional area of gas ducts and thus the dimensions of the boiler plant.

In the combustion of high-sulphur fuel oils, the metal of heating surfaces in the gas path may be subjected to corrosion under appropriate conditions. The principal method for preventing high-temperature corrosion is to maintain the temperature of the heating surface walls below 600°C. Low-temperature gas corrosion is prevented by raising the temperature of air at the inlet to the air heater and

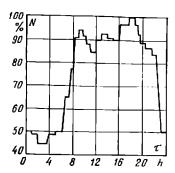


Fig. 22.3. Daily load curve of a power system

by using corrosion-resisting materials and protective coatings. These methods have been described in Ch. 19.

Power engineering is the sole branch of national economy where the final product is not stored, but consumed completely just as it is produced. This specific feature defines the interrelation between the operating modes of energy-producing systems and energy consumers. This determines one of the principal problems: organization of the optimal conditions for operation of electric power systems, power stations and their equipment.

The total power developed at a particular moment by all plants of a system is called the system load. Figure 22.3 shows a curve of daily electric load of a power system which includes all its consumers: industrial and agricultural plants, electric transport, illumination and appliance loads. Peaks of the load curve are due to uneven operation of industrial enterprises, transport, illumination, etc.

As regards the loading coefficient and the utilization of installed power of monobloc units, they may operate to supply a base load, i.e. to operate at their rated capacity for 6 000-7 500 hours per year ($\tau_{max} = 6 000-7 500 \text{ h}$), a semi-peak load with $\tau_{max} = 2 000-4 000 \text{ h}$, or a peak load with τ_{max} less than 2 000 h. Large monobloc units for supercritical pressures usually supply a base load and are not suited well for load variations. On the

other hand, manoeuvring units, i.e. those for supplying semi-peak and peak loads, are characterized by a low time of utilization of the installed power and an elevated speed of start-up. Subcritical-pressure plants are more suitable for supplying semi-peak and peak loads than those operating at supercritical pressures.

Monobloc units for supercritical pressures require a longer time for boiler firing and heating up of pipelines and greater heat consumption for start-up. They are less convenient for control. Their valves and fitting are worn more rapidly, especially with frequent start-ups, and the metal is more liable to develop high thermal stresses owing to often load variations at the high pressure.

For the European territory of the USSR, subcritical-pressure units of the capacities up to 500 MW are recommended for suppling semi-peak and peak loads. Steam reheating is often employed in order to avoid erosion of turbine blades in the last stages, especially at variable load conditions, though this makes the plant more expensive.

For the reliable operation of pearlitic steels under semi-peak and peak load conditions, the temperature of live and reheated steam can be taken equal to 530-535°C. To prevent erosion in the last stages of the turbine, the lower limit of superheated steam temperature is 480-500°C. As a result, the steam temperature of 510°/510°C turns out to be the most preferable from the considerations of minimizing the metal consumption in the superheater and decreasing the time and cost of start-ups.

When selecting the initial pressure of steam, it should be remembered that pressures of 13 MPa and 16 MPa ensure roughly the same economic efficiency characteristics. The manoeuvrability of the power units at these pressures is also roughly the same. In view of the positive experience of operation of steam-turbine plants at a pressure of 13 MPa, it is advisable to

design manoeuvring power units for the same initial pressure.

The experience of operation of highcapacity high-pressure drum-type boilers in the power units for 100-200 MW has revealed that cracks of corrosionlatigue origin form in the boiler drum, which appeared due to non-uniform temperature fields at variable load conditions. In the boiler drums of more powerful units (500 MW), where the drum has larger dimensions and thicker walls, the danger of high temperature non-uniformity and temperature stresses is higher, especially at varying operating modes in which the equipment is shut down and started up too frequently. In once-through boilers there are no such heavy metal elements as the drum; they are lighter and can be shut down and started up more quickly. For this reason, oncethrough boilers are recommended for power units which supply semi-peak and peak loads.

A manoeuvrable steam boiler should be compact and the mass of metal and other materials in it should be as low as possible. With a lower mass, less time is needed for a change from one thermal state (mode) to another. This condition is more easily met in fuel oil-fired boilers which also have a wider range of operating loads in which fuel combustion can be efficient and stable.

Since peak-load boilers are in operation only a limited number of hours, they are made less expensive by reducing the consumption of metal for the low-temperature heating surfaces; in this case the temperature of waste gases is taken to be 160-180°C.

For more convenient maintenance, all heating surfaces of a manoeuvring boiler should be drainable, because of which the convective elements are made with the horizontal arrangement of tubes.

The working fluid in once-through manoeuvring boilers has an elevated mass velocity which ensures proper cooling in a wider range of loads and reliable operation of a boiler at low loads. Recirculation of the working fluid in water walls by means of recirculation pumps can also serve the same purpose.

22.2. Characteristics of Modern Steam Boilers

Most supercritical-pressure oncethrough boilers at thermal power stations operate in monobloc units of a capacity of 300 MW or more. There are a number of once-through boilers for subcritical pressures which are installed in 150-200-MW monobloc units, but their manufacture has been discontinued.

The high-capacity once-through boilers produced in the sixties were mainly of the two-housing scheme, i.e. in the form of two symmetrical housings constituting a double-bloc together with the turbine. A double-bloc can operate at a load half of the nominal with one of the boiler housings shut down. The experience of operation of the double-bloc plants has revealed no essential advantages over the single-housing type. On the other hand, their disadvantages (more intricate layout, greater number of flows of the working fluid, and greater number of valves and fittings) make them less convenient in exploitation. Their manufacture has been abandoned in favour of single-housing steam boilers for monobloc units.

Single-housing boilers for 300-MW monoblocs have a prismatic (open) furnace without constriction, which makes it possible to reduce the average heating intensity of water walls in the flame core zone to a safe level. The same object is achieved by the recirculation of combustion products which are taken at a relatively low temperature from the convective gas duct (usually downstream of the economizer) and supplied by the recirculation fan into the furnace. This decreases the heat absorption in the lower radiation section, which is favourable for stabilizing temperature conditions of the water walls and decreasing high-temperature corrosion. Fuel is burned at the minimal excess air ratio which is effective for combatting low-temporature corrosion and preventing fouling of the heating surfaces.

Recently, steam boilers with gastight enclosures have found wide application. The principle of gas tightness can be more easily realized when the water walls are made from vertical sections with a single-pass flow of the working fluid.

Gas-tight water walls substantially increase the economic efficiency and reliability of boiler plants. The principal advantages of gas-tight boilers are as follows: there are no air inleakages into the furnace and gas ducts (and therefore, q_2 is lower); the boiler plant consumes less auxiliary power for the transport of air and combustion products; combustion can be organized under the optimal conditions with the least excess air ratio, which low-temperature corrosion prevents and fouling of the heating surfaces (in combustion of high-sulphur fuel oil, this also prevents high-temperature corrosion); heavy refractory ting is replaced by light heat insula-(this decreases heat losses, shortens the time for start-up and shutdown, and reduces the mass of the boiler structure and foundation); slag and soot can be more easily removed from the furnace by water washing of the water walls without risk to damage the lining.

Supercharging places a number of additional requirements on the boiler design: the single-housing schome should be preferred so as to diminish the unit surface area of expensive gastight walls; all-welded gas-tight water walls serve as the enclosure of the furnace and gas ducts; the number of individually controlled flows of the working fluid is smaller, and therefore, there are fewer points of passage of the tubes through gas-tight walls which must be properly sealed; the furnace chamber has a narrower front, but greater depth and height (to reduce

the number of successively connected water wall sections which must be welded together by their sides); the temperature difference between the welded sections is decreased due to recirculation and bypassing of working fluid in water walls.

In supercharged boilers of supercritical pressures, the superheater tubes are passed from the furnace through welded roof sections, with a second roof being provided on the furnace. The convective reheater banks are arranged in the down-take gas duct horizontally; the probable effect of non-uniform temperature distribution is minimized by sectionalizing the reheater across the gas duct width into autonomous parallel flows. All sides of the convective shaft are formed by the gas-tight water walls of the economizer and superheater.

The economizer, live-steam reheater and regenerative air heater in supercharged boilers are essentially of the same design as in balanced-draft boilers. The boiler structure has additional beam belts spaced roughly 3 m apart to absorb the strosses appearing due to supercharging.

In modern high-capacity boiler plants, the load-carrying boiler structure is mostly combined with the building structures (see Sec. 21.1). which gives a noticeable saving in metal. For instance, the metal saving for a gas and fuel-oil fired boiler type TGMP-204 for an 800-MW monobloc is around 1 500 This unit design is employed in new monobloc boilers, of capacities of 500, 800 and 1 200 MW, both gas and fuel-oil fired and those fired on pulverized coal.

An example of high-capacity single-housing boilers for supercritical steam parameters is shown in Fig. 22.4. The boiler type P-67 for 800-MW monobloc units is fired on Kansko-Achinsky coal; its steam-generating capacity is $D=2\,650$ t/h and steam parameters: 25.5 MPa, 545°/545°C. It has the T-shaped layout with the boiler structure suspended from the boiler

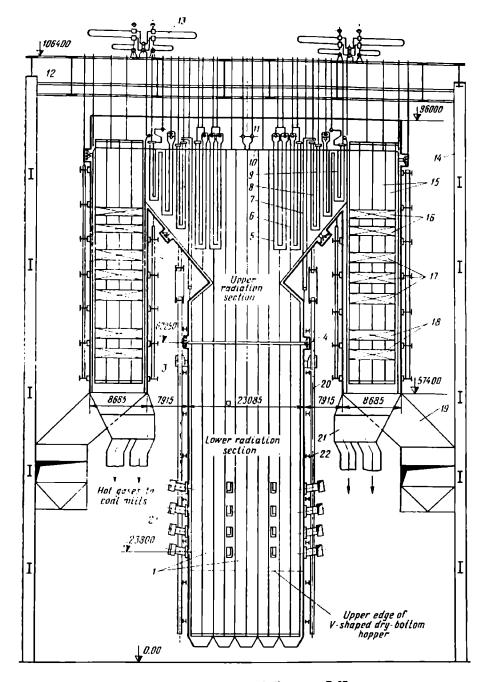


Fig. 22.4. Steam boiler type P-67

Fig. 22.3. Steam sporter type 1-07

1-all-welded water walls in furnace; 2-burners; 3-recirculated gases; 4-expansion joints (parting joints);
5-ist platen superheater; 6-2nd platen superheater; 7-slag screen; 8-3rd platen superheater (the last section of supercritical-pressure superheater); 8-the last bank of reheater; 10-roof tubes; 11-'bot box';
12-main heams; 13-live-steam reheater; 14-boiler room columns; 15-all-welded water walls in convective furnace shaft; 16-ist stage of convective supercritical-pressure superheater; 17-tst reheater stage;
18-economizer; 19-gas duct for the removal of combustion gases to tubular air heater; 20-tubes suspended from intermediate beams ('hoi' suspensions); 21-duct for removal of drying gases to pulverizing mills;
22-rigidity beits

room building structure. The boiler has gas-tight water walls, but operates with balanced draft. The furnace is of the open dry-bottom type, square in plan and has four tiers of tangential burners (two vertical groups of burners on each furnace wall).

In view of the high explosibility of the pulverized Kansko-Achinsky coal and the high fuel consumption by the boiler, the pulverizing system of the boiler is provided with gas drying of fuel and mill fans which inject coarse coal dust into the furnace.

In order to minimize slagging of water walls, fuel combustion is organized at a low temperature (less than 1 260°C) and a low average heat release rate to water walls. For the same purpose, part of the combustion products (roughly 20%) is taken from the horizontal gas duct at a temperature of 670°C and recirculated through a water-walled gas conduit to the burners behind the gas drier. The tangential burners form a vertical turbulent flame in the furnace centre, so that the direct contact of the flame with the water walls is minimized.

The heating surfaces behind the furnace are of the platen type, which minimizes the probability of tubes slagging. Slagging is also prevented by recirculating another 10% of the combustion products through nozzles into the top portion of the furnace so as to maintain the gas temperature before the platens at a level below 1 025°C. The recycled gases are taken off by two exhaust fans behind the economizer at a temperature of 350°C.

The boiler has two symmetrical convective shafts each of the cross-sectional size 23 085 mm × 8 685 mm. Each shaft is divided vertically into two portions by a gas channel 2 300 mm wide formed by two gas-tight walls which carry no heating surfaces, i.e. there are essentially four convective shafts. Combustion products are taken off at a temperature of around 800°C at the entry to convective shafts and move along the gas channels to pulverizing fans (Fig. 22.4, item 21).

A tubular air heater is mounted in a separate bay of the main building.

The water-steam path of the boiler has two circuits with non-mixing separately controlled flows, which are symmetrical about the vertical axis of the boiler. The boiler is provided with an overhead travelling crane of a load-carrying capacity of 30 t and with goods and passenger lifts.

Figure 22.5 shows a gas and fuel-oil fired boiler type TGMP-1202 of suspended design for 1 200-MW monobloc units. It is of the single-housing layout, which makes it possible to organize single-pass ascending motion of the working fluid in the furnace water walls at a mass flow rate wo ≈ $\approx 2000 \text{ kg/(m}^2\text{s})$. The boiler is supercharged. The furnace is 31 280 mm 10 420 mm in plan and carries membrane sections of unified width. The enclosing sections of the convective and connecting gas ducts are also unified. The furnace has a rather large depth which allows free development of the flame without touching water walls. Firing-up of the boiler is done by using the fuel oil burners. In order to diminish the firing-up load (to $0.15D_{r}$) and increase the reliability of all-welded water walls, the boiler is provided with a water recirculating system which operates when the load is less than 40-50% of D_r . The burners are arranged in two fronts and three tiers. Combustion gases are taken off behind the economizer and recirculated into the top and bottom portion of the furnace. The working fluid moves in the furnace water walls by the single-pass mode. There are two firing-up units, one for each flow. The supercritical-pressure superheater (D = 3.950 t/h) is arranged in the horizontal duct and comprises platen sections and two convective stages. The supercritical-pressure steam path has two water injection points: one before the platens (5% of the total water flow rate) and the other before the last stage of the convective superheater (2%).

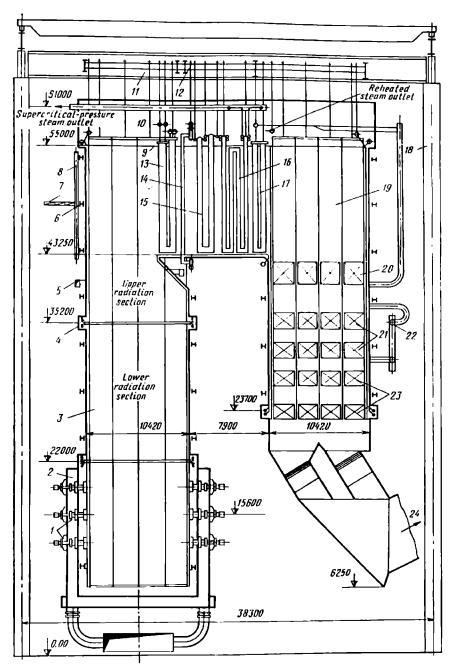


Fig. 22.5. Steam boiler type TGMP-1202

Fig. 24.0. Seean Botter type 1 City 1202 |
1—gas-fuel oil burners; 2—air box; 3—furnace water walls; 4—borizontal parting joint; 5—nozzles; 6—
rigidity helt; 7—horizontal truss; 8—suspended tubes; 9—roof water wall; 10—'hot box'; 11—main beam;
12—distributing celling; 13—superheater platens; 14—five-tube slag screen; 15—1st stage of convective superheater; 16—2nd singe of convective superheater; 17—the last reheater bank; 16—bolier room columns; 19—gas-tight water walls in convective furnace shaft; 20—reheater stage; 21—control stage of reheater; 22—control bypassing valve; 23—economizer; 24—combustion products to regenerative air heater

The low-pressure path has a control, an intermediate and an outlet bank. At the rated capacity and fuel oil firing, roughly 30% of steam pass through the control bank and the remaining 70% are bypassed. The two steam flows are then mixed in a header and fed into the intermediate and the outlet bank.

The economizer has two tube banks. There are four regenerative air heaters of 12.9-m diameter.

The semi-peak once-through boiler type TMP-501 of a steam-generating capacity 1 800 t/h (14 MPa, 515°/515°C) operates in a monobloc with a 500-MW steam turbine (Fig. 22.6). It has the Π-shaped layout. The open-type prismatic furnace has weakly inclined water wall tubes coiled all over its walls in the lower radiation section; this prevents temperature stratification in the zone of intensive heating of the working fluid which might occur at low loads and transient regimes, i.e. when some walls are heated unevenly. In the less heated upper radiation section, water walls are made in

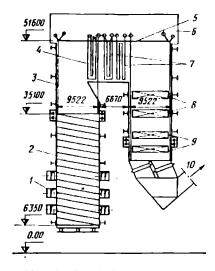


Fig. 22.6. Semi-peak hoiler type TMP-501 (D = 1800 t/h, p = 14 MPa, 515°/515°C)

1-burner; 2-lower radiation section; 3-upper radiation section; 4-superheater platens; 3-roof superheater; 6-hot box; 7-convective superheaters; 8-reheater; 9-economizer; 10-to regenerative air heater

the form of vertical sections. Since there are no substantial temperature differences of the working fluid in parallel tubes, water walls are of the all-welded design and the boiler is gas-tight.

Since the superheated steam temperature is relatively low, all heating surfaces of the boiler are made from poarlitic stool which operates at variable temperatures more reliably than austenitic steel. The headers have a low wall thickness (less than 36 mm) so as to prevent substantial temperature stresses and thus increase their reliability. This has become possible because of the moderate steam pressure in the boiler. The temperature of live steam is controlled by water injection and that of reheated steam by the recirculation of combustion products to the furnace burners and, in addition, by water injection at the full!load.

In view of the high risk of corrosion at low loads and at frequent starting-up and shutting-down, the boiler is provided with two regenerative air heaters with a porcelain packing in the cold sections and a steam air heater which ensures a high temperature of air preheating in all operating regimes. The boiler is suspended from the main beams of the boiler room building.

Figure 22.7 shows the steam boiler for 1 300-MW monobloc unit of the Gavin power station in the USA. It is of the single-housing type with IIshaped layout. The open-type furnace for firing pulverized coal has oppositely arranged burners and a dry-bottom hopper. Its width, depth and height are respectively 34.4 m, 15.8 m and 61 m. The boiler has 14 roller mills, with seven mills mounted on each side. The rated capacity is covered by the operation of 10-12 mills, the remaining being the reserve. Pulverized fuel is dried by air and blown directly into the burners. Capital repairs of the roller mills are carried out once in two years. Fuel is supplied to the mills by closed belt conveyors provided with automatic scales to

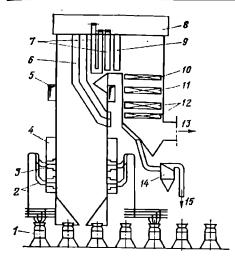


Fig. 22.7. Once-through boiler (D=4438 t/h, p=27.3 MPa, $543^{\circ}/538^{\circ}$ C) 1—pulverizing mills: \mathcal{E} —pulverized-coal ducts; 3—burner; ℓ —secondary air box; ℓ —box for recirculated gases; ℓ —platens; ℓ —the last superheater stage; ℓ —thot box'; ℓ and ℓ 1—reheaters; ℓ 1—to regenerative air heater; ℓ 2—conomizer; ℓ 3—to regenerative air heater; ℓ 4—dust collection from recirculated gases; ℓ 5—to recirculating gas fan

measure fuel consumption. Each mill is connected with the burners by eight dust pipelines of an appreciable length which also compensate for the burner displacement owing to temperature expansion of the water walls. The boiler has 112 burners arranged in four tiers on the front and rear walls of the furnace (the burners are enclosed by secondary-air boxes).

The furnace water walls are made in the form of all-welded sections with vertical tubes and ascending motion of the working fluid. There is a single parting joint along the furnace height. Three platens are mounted in the furnace top. The boiler is provided with a gas recirculation system which returns the combustion products into the furnace in two points: a smaller portion through the tubes of the dry-bottom hopper (not shown in the figure) and a greater portion, through the boxes at the furnace top.

The boiler is supercharged, the surplus pressure being developed by three air blowers. There are no reserve

blowers. The boiler structure is suspended on long ties of a diameter of 120 mm and spring-biased supports. The heating surfaces can be blown by fixed and retractable nozzles up to 17 m long, the blow-down system being switched on automatically twice a day.

Drum-type boilers are made for subcritical steam pressures. The drum-type boilers employed at Soviet thermal power stations are of the natural circulation type and mostly with balanced draft. In other countries, forced-circulation drum-type boilers are also in use.

The drum-type boiler type TPE-211 with natural circulation (Fig. 22.8) has been developed on the basis of the similar boiler type TP-100 which has found wide application. It is intended for the combustion of coals (dry-bottom furnace) and natural gas. It is of the T-shaped layout, with the furnace being arranged in the central uprising shaft and the low-temperature convective heating surfaces, in two vertical convective shafts symmetrically on the left and right of the furnace. The convective shafts are connected with the furnace by horizontal gas ducts in which the high-temperature convective superheater sections are mounted.

The open-type furnace of the prismatic shape is oriented by its longer side across the main building and carries evaporating water walls. It is vertically divided by a platen into two symmetrical furnace chambers. At the bottom, the platen together with opposite water walls forms two cold-bottom hoppers. Each section of the furnace is provided with pulverized coal and gas burners arranged in two tiers on its side walls. The boiler drum is mounted transversely to the furnace front; it is made of steel Grade 16GNMA (0.16C-Mn-Nihigh quality) and has an internal diameter of 1 800 mm and wall thickness 112 mm.

Superheating of live steam is effected in a superheater section on the

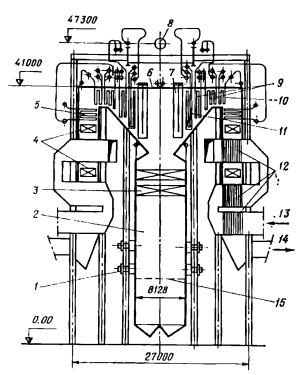


Fig. 22.8. Drum-type boiler TPE-211 (D = 670 t/h, p = 14 MPa)

1 - burners; 2 - furnace; 1 - radiant superheater section; 4 - cconomizer; 5 - control stage of superheater; 6 - roof superheater; 7 - platen; 8 drum; 9 - reheater; 10 and 11 - convective superheater sections; 12 air heater; 13 - cold air in; 14 waste gases; 15 - upper end of drybottom hopper

furnace roof and in the radiant superheater located in the top portion of the furnace under the constriction above the water wall tubes. The convective superheater and reheater are arranged almost completely in the horizontal gas ducts. Steam for reheating is supplied from the turbine in two flows, each of which is separated into two flows in a control bypass valve and can be directed partially into an additional (control) reheater surface and partially around it into the convective banks. The temperature of high-pressure superheated steam is controlled by the injection of boiler condensate. The economizer is arranged in the downtake shaft together with its headers. The air heater is of the tubular three-pass type.

In conventional boiler designs where water walls are arranged on the enclosing walls of the furnace, the dimensions of the boiler plant may be too large. Their dimensions can be decreased by applying forced-flow tur-

bulent furnaces or by intensifying heat transfer in the heating surfaces by using finned tubes. An example of a small-size boiler plant is the boiler type TGM-444 developed at the Centboiler and turbine (Fig. 22.9). It is essentially a gastight supercharged drum-type boiler. The furnace has two chambers: a turbulent primary furnace at the bottom and a furnace chamber with haretube water walls above it. The total surface area of the heating surfaces is enlarged by adding two finnedtube platens which are arranged perpendicular to the furnace front and divide the furnace along its whole height into three sections. All water walls of the furnace are connected into the natural-circulation circuits. Instead of a large number of downtake tubes, the boiler has four water-descending stand-pipes of 426-mm diameter and 36-mm wall thickness.

The front wall of the furnace carries six straight-flow gas and fuel oil

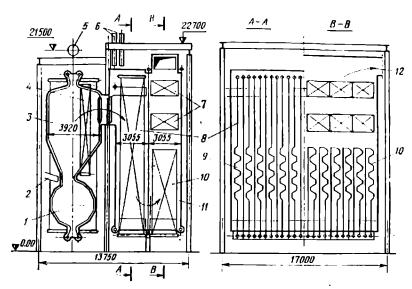


Fig. 22.9. Small-size gas and fuel oil fired boiler type TGM-444 ($D=500~{\rm t/h},~p=14~{\rm MPa},~560^{\circ}{\rm C}$)

1—cyclone primary furnace; 2—burner; 3—cooling chamber; 4—furnace platen; 5—drum; 6—external cyclones; 7—economizer; 8—platen; 9—bent-tube bank; 10—convective superheater platens; 11—ascending gas duct; 12—to regenerative air heater

burners (two in each furnace section). In the zone of the combustion products exit from the furnace there are 12 vertical all-welded single-pass superheater platens. The heat release rate per unit volume of the primary furnace is roughly 2 MW/m³ and for the whole furnace 0.7 MW/m³.

A horizontal gas duct and two vertical connected gas ducts are mounted behind the furnace chamber, all enclosed by all-welded water walls. The vertical shafts carry convective platen sections of the superheater. At the outlet from the ascending gas duct, combustion products pass through an economizer and, finally, through a regenerative air heater (one for the whole boiler plant). All heating surfaces of the boiler are drainable.

The boiler offers the following advantages: it has substantially smaller dimensions (by 30-40%) compared

with conventional boilers and by 25-30% lower mass of the metal; the volume of construction work for the main building of the power station, including the boiler, can be decreased by 50-66% and the labour consumption for the boiler manufacture and mounting by 30-40%. The boiler is universal as regards its thermal characteristics and manufacture. The latter is achieved by developing a standard furnace module of a length M =4 480 mm and diameter 3 960 mm. A number of modules can be combined to form the furnace for boilers of various capacities. For instance, the boiler described (D = 500 t/h) requires three modules, i.e. the length of the primary furnace (width across the front) is: $L = 3M = 3 \times 4480 =$ 13 440 mm. Small-size forced boilers are in the stage of industrial development.



STEAM BOILER OPERATION

23.1. Operating Conditions and Characteristics

The principal object of boiler operation is to satisfy the current power demand according to the load curve and ensure the highest service life and highest efficiency of boiler plant.

The load curve of a power station is mostly non-uniform which is determined by variations of energy consumption by the consumers. There are daily, weekly and seasonal load curves. As a rule, a load curve has peaks in the morning and in the evening and drops down at daytime and especially at night. Load drops are also observed on non-working days, especially at night. The seasonal load curve mostly has a maximum in autumn and winter. Energy-producing equipment may operate in the base, semi-peak or peak regime (mode). In a base regime, the load is essentially constant, though certain variations are allowed in this regime and even shut-down to reserve on non-working days. In semi-peak regime, the station equipment is shut down to reserve in the nighttime and all the days of rest. In peak regime, the equipment is in operation only during peaks of the load.

The following principal characteristics are adopted to describe the operating conditions of boilers:

(a) the net efficiency of a boiler at rated load and the average efficiency for a particular operating period;

(b) the operation factor which is essentially the ratio of the actual operating time of a boiler, τ_{op} , to the calender time τ_{eal} of the period considered (month or year);

(c) the availability factor, i.e. the ratio of the total time during which a boiler was in operation and reserve to the calender time;

(d) the capacity factor which is the ratio of the steam generation during τ_{op} to the probable steam generation during τ_{cal} if the boiler operated at the rated steam-generating capacity; and

(e) the average and maximum time of a campaign or, using the terminology from the reliability theory, the operating time to failure (a failure is understood as an event that disturbs the operating ability of a boiler).

Boilers operating in base regimes should have the highest values of the indicated characteristics. It should be clear that such high indices cannot be attained by the boilers operating in semi-peak and peak regimes.

Since monobloc units gradually replace older equipment in thermal power stations of power systems, some of them, including those of supercritical pressures, are employed increasingly for supplying the variable portion of the load curve. In that respect, the manoeuvrability of boiler equipment is a characteristic that grows in importance.

The concept of manoeuvrability of monobloc units includes: the characteristics which determine the range of operating loads of a unit; the startup-shutdown characteristics; the dynamic properties; the characteristics at sudden sheddings and surges of the load.

The conditions of operation of boiler equipment at various loads are characterized by the load control range and the range of allowable loads. In the load control range, the automatic con-

trol system of a boiler should quickly respond to load variations without interference of the personnel. In this range, it is not allowed to change the composition of the operating auxiliary equipment, the number of boiler burners and the adjustment of automatic regulators. The concept of the allowable load range includes, in addition, the region of loads from the lower limit of the control range to the lowest capacity at which the boiler can operate steadily. In this range, the load varies relatively slowly according to the load-dispatching curve and it is allowed to vary the composition of the auxiliary equipment, change the number of burners, and 'brighten' the pulverized-coal flame by startup fuel in order to stabilize combustion. It is also allowed to vary the adjustment or switch off some of the automatic regulators. The lowest capacity of a boiler is determined by the reliability of temperature and hydraulic conditions of the water walls and the stability of fuel combustion and discharge of molten slag from the slaghole.

Regimes of operation at any load with insignificant variations of steam parameters are called steady. In contrast, regimes characterized by load variations and deviations of steam parameters due to internal or external disturbances are called unsteady. Internal disturbances of an operating regime are caused by variations of one or more boiler inlet parameters (flow rate or temperature of feed water, fuel consumption, air flow rate, etc.). External disturbances are caused by variations in the external conditions (such as the pressure of steam in steam main, the load of the turbo-generator. the degree of opening of the startupshutdown device, etc.). An important characteristic of a boiler is its ability to change the load quickly, which is often called the acceleration characteristic. It is determined mainly by the dynamic properties of a boiler, i.e. by its response to regime disturbances.

The characteristics of manoeuvrability of the boiler equipment are determined largely by startup-shutdown regimes. The principal types of boiler shut-down are as follows:

- (1) shut-down to reserve or for some types of repair work which require no cooling of the boiler and steam pipelines:
- (2) shut-down for repairs with cooling of the whole boiler plant or a particular element;
- (3) emergency shut-down; it is carried out by the prescribed technology which is determined by the cause of shut-down and the possibility of restarting.

The firing regime of a boiler includes heating the water-steam and gas-air paths, after which the process of steam generation begins and steam parameters are raised to the specified values. For monobloc units, boiler firing is a part of the starting-up regime of a unit. Firing a boiler is considered to be completed when the steamgenerating capacity and steam parameters have been raised to the specified values so as to 'push' the turbine rotor and accelerate its rotation. This regime is also called the starting-up regime. The firing and starting-up regimes heavily dopend on the original thermal state of the equipment. A boiler can be started from a hot state, a warm state or a cold state. For monobloc unit boilers, these regimes are distinguished roughly by the time of the preceding idle time (respectively 6-10 h, from 6-10 h to 70-90 h, and more than 70-90 h). For conventional boilers (not monobloc) the difference between the first and second group is determined by the time of the preceding idle time (6-10 h or more) and between the second and third group, by the loss of gauge pressure in the boiler and the drop in the temperature of the hottest elements down to 150°C. For monobloc boilers, in addition, the starting-up regime from the state of hot reserve is known.

23.2. Steady Regimes of Boiler Operation

Steady conditions of boiler operation have been considered in many preceding chapters, so that the present chapter will give only some additional data. In the general case, boiler operation is carried out according to a regime chart which gives the principal characteristics of the regime obtained by the results of boiler tests. Modern monobloc units are equipped with computer means which make it possible to determine the current or average (for any given period, such as a shift) efficiency characteristics and the characteristics of the working fluid in the water-steam path of the boiler. This information is used for the optimization of boiler operation. In addition, a number of automatic regulators maintain the process parameters at optimal values (for instance, the air regulator maintains the specified concentration of oxygen in flue gases). For these devices the regime chart gives the extreme variations of the variables, above which the personnel should correct the operation of the controllers or change to manual operation.

Controlling the efficiency of combustion process consists in maintaining the optimal excess air ratio in the furnace (α_i) and in distributing the air flow between the burners in accordance with fuel distribution. The optimal α_i is found by minimizing the heat losses (see Ch. 6). Uniform distribution of fuel and air between the boiler burners is favourable for decreasing the maximum temperatures of water-wall tubes, decreasing temperature stratification of flue gases at the furnace outlet, and preventing slagging of the heating surfaces.

It is essential to hold the temperature of tube walls of the heating surfaces within the limits allowed for their metal (see Ch. 10). Direct measurement of temperature of the tube metal can only be carried out in boiler tests. In boiler operation, these tem-

peratures are controlled by maintaining the specified temperatures of the working fluid at the outlet from particular heating surfaces. In some cases, these temperatures may exceed the specified values. For instance, in oncethrough boilers this may occur on a decrease in the flow rate of feed water or an increase in the flow rate of spray water (the total or in some spray nozzles). In any boiler, this also can result from improper conditions in the furnace. For instance, if the combustion core in the boiler furnace is too high, it may raise the steam temperature behind some stages of the superheater, especially behind the unprotected superheater sections. Non-uniform distribution of fuel between the burners may lead to an increased temperature of one of gas flows and to increased maldistribution of heat in some heating surfaces. It is clear that in once-through boilers it is essential to redistribute continuously the flow rate of feed water and the flow rates of fuel between the burner tiers and furnace sides so as to ensure reliable operation of the heating surfaces. In drum-type boilers, an additional variable is the water level in the drum. An excessively high water level can lead to carry-over of water droplets which can impair the steam quality and form internal deposits (scale) in superheater tubes. This may result in a higher temperature of the tube metal and even in burnthrough.

The temperature conditions of the heating surfaces are also impaired if some tubes have leaks or bursts. Tube-burst indicators of the acoustic type are finding increasing use for detecting bursts and leaks in boiler tubes. If the temperature behind particular heating surfaces has increased above the allowable level or if a tube burst has been detected, the boiler is shut down (see Sec. 23.5). In addition to the current control of temperature conditions in a boiler, the state of the metal of the heating surfaces, headers and steam pipelines operating at tem-

peratures above 450°C is checked periodically by measuring their dimensions or by cutting metal specimens from the tubes for metallographic analysis.

An important task of the hoiler personnel is to organize the boiler operation so as to minimize low-temperature corrosion in the tail heating surfaces and gas ducts. As has been given in Ch. 16, it is essential for the purpose to burn high-sulphur fuel oil at the least excess air ratio (1.02-1.03) and maintain the temperature of the preheated air at a level of 70°C before a tubular air heater or 60°C before regenerative air heaters. Combustion of fuel oil at a low excess air ratio involves certain difficulties, in particular, the furnace should be perfectly tight, fuel and air should be uniformly distributed between the burners, the plant should be provided with oxygen meters and smoke density indicators. If these conditions are not observed, soot will be deposited on the heating surfaces (its ignition may cause damage to the tubes) and ejected into the atmosphere.

Steady regimes of the boiler operation may be different depending on the load. Variations of a particular steam parameter or regime characteristic as a function of load are described by what is called the static characteristic. As follows from what has been said in Ch. 18, an increase in the load decreases the fraction of heat absorbed by radiation and increases that absorbed by convection. In a drum-type boiler, this leads to an increase in the temperatures of live and reheated steam, water behind the economizer, and hot air. In once-through boilers where no distinct boundary is observed between the steam-generating and superheating sections, the temperature of live steam may remain constant. Typical static characteristics of live and reheated steam in a once-through and drum-type boiler are shown in Fig. 23.1.

With an increase of load, the temperature of flue gases at the furnace outlet increases, resulting in a greater

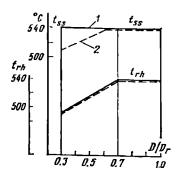


Fig. 23.1. Static characteristics of fresh and reheated steam for (1) once-through and (2) drum-type boiler

volume of combustion products and higher temperature gradients in the convective heating surfaces whose heat absorption thus increases somewhat. Since the flue gases have a higher temperature and the excess air ratio is unchanged, the loss q_2 increases. In some cases, the boiler furnace is run at an elevated excess air ratio to form appropriate furnace conditions or maintain the specified temperature of steam at reduced loads. In this case the loss q₂ may even increase at reduced loads. The loss of heat due to incomplete combustion, q_3 , is usually very low. The loss with unburned carbon, q_4 , first decreases with increasing load, due to a higher temperature level of combustion, but then can rise due to a shorter time of fuel particles presence in the furnace. The loss of heat to the surroundings, q_5 , decreases with an increase in load. It can be concluded from the foregoing that the relationship between the boiler efficiency and load is influenced by a number of factors: boiler design, fuel grade, air conditions, air inleakage, etc. The highest efficiency can be obtained at a load somewhat below the rated load [near (0.8-0.9) D_r in the combustion of solid fuels and (0.6-0.7) D_r in the combustion of fuel oil].

A change in fuel consumption on a change of load is determined by the

ratio of balance equations:

$$\frac{B_x}{B_r} = \frac{[D_{sh}^x (i_{ss} - i_{fw})^x + D_{rh}^x (i_{rh}^r - i_{rh}^r)^x] \eta_b^r}{[D_{sh}^r (i_{ss}^r - i_{fw})^r + D_{rh}^r (i_{rh}^r - i_{rh}^r)^r] \eta_b^x}$$
(23.1)

where D_{sh} , D_{rh} are the flow rates of live and reheated steam, i_{ss} and i_{rh}^{r} are respectively their enthalpies, i_{fw} and i_{rh}^{r} are the enthalpies of feed water and steam at the reheater inlet and the subscripts 'r' and 'x' refer to the rated and reduced load.

If load is reduced at constant rated parameters of steam, i_{fw} and i'_{rh} will decrease with only a slight change of η_b , so that the fuel consumption will change less substantially than the boiler load.

As has been established by boiler tests and by experience of boiler operation, the lower limit of the load control range is as follows: 40-50% of the rated load in the combustion of fuel oils, gas and high-volatile coals; 50-60% for lean coals; and 60-75% for slagging-bottom boilers. The lowest load for stable operation of a boiler is usually 30-40% of the rated load.

23.3. Unsteady Regimes of Operation Within Allowable Loads

Let us consider the principal unsteady processes which can occur in boilers on the typical disturbances of their operating regimes. According to the laws of conservation of mass and energy, we can write the equations of material and heat balance:

$$G_a - D_d = \frac{\{d_i^1\}}{d\tau} (G_{in}) \dots \} (23.2)$$

$$\eta_b Q_s - Q_d = \frac{d!}{d\tau} (Q_{in}) \dots (23.3)$$

where G_s and Q_s are the quantities of water and heat supplied to the boiler, D_d and Q_d are the quantities of steam and heat delivered from it, and G_{in} are the quantities of water and heat contained in the boiler.

The term G_{in} is the mass of water and steam contained in the boiler (it may be called mass filling). The term

 Q_{in} is the quantity of heat accumulated in the boiler:

$$Q_{ln} = \sum G_m c_m t_m + \sum G_w c_w t_w + \sum D_s c_s t_s \dots$$
 (23.4)

where c is specific heat and the subscripts 'm', 'w' and 's' stand respectively for 'metal', 'water' and 'steam'.

Specific processes occurring in the drum-type and once-through boilers are determined by the differences between these boiler types: the position of the boundaries of the steamgenerating portion, its filling mass, and capacity of heat accumulation. It should be recalled, in particular, that the steam-generating portion in the drum-type boilers has the boundaries determined by the drum, whereas in the once-through boilers these boundaries are indeterminate. Since the drum-type boiler has a multiple circulation, the steam content at the outlet from the wall tube system is low and the mass filling is respectively high. On the other hand, in oncethrough boilers, the working fluid circulates in a single pass, so that the steam content increases substantially and the mass filling is appreciably lower than in the former case. Besides, water-wall tubes in the drum-type boilers have a larger diameter (to ensure proper circulation) than in once-through boilers, which again increases their mass filling and the heat accumulated by the metal. The drum is also an element that accumulates much heat, so that the terms $d (G_{in})/d\tau$ and $d\left(Q_{in}\right)/d au$ for the drum-type boilers are substantially higher.

Such are the principal differences in unsteady processes occurring in these two principal types of boiler*.

Let us analyse a regime in which the fuel consumption B (and therefore, heat release in the furnace) increases at a constant flow rate of feed water G_{fw} . In a once-through boiler

^{*} A more detailed discussion of unsteady processes and their mathematical description can be found in [56, 61], etc.

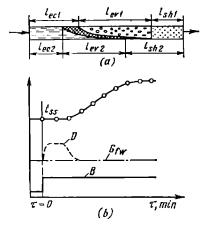


Fig. 23.2. Variations of operating conditions of a once-through boiler with increasing fuel consumption

(a) displacement of boundaries of the steam-generating portion; (b) variations of the flow rate and temperature of steam

(Fig. 23.2), the surface area of tubes required to heat water to the boiling temperature decreases in this case, and therefore, the boundary of the economizer portion displaces in the direction against the water flow (from l_{ec1} to l_{ec2}). The heating surface area of the steam-generating portion also decreases (from l_{ev1} to l_{ev2}) and, besides, its beginning displaces. Therefore, the point in which steam generation is completed shifts against the flow more substantially than the boundary of the economizer portion. Thus, the mass filling of the boiler decreases (see the hatched area in Fig. 23.2a), while the superheating portion of the boiler increases (from l_{sh1} to l_{sh2}). These boundaries displace gradually. First, the boundary of the beginning of steam generation is shifted and, since part of the water contained in the boiler is evaporated, the steam flow rate D exceeds for a certain time the flow rate of water G_{tw} . During this period, the ratio B/D remains constant and the temperature of live steam t_{ss} is not changed (Fig. 23.2b). Further, as the boundary of the end of the steam-generating portion is shifted and the heat accumulated by

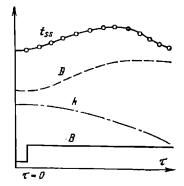


Fig. 23.3. Variations of the operating conditions of a drum-type boiler with increasing fuel consumption

the working fluid and metal increases, t_{ss} becomes higher. The processes occurring in a supercritical-pressure boiler can be presented in a similar manner if the zone of the highest heat capacity of the fluid is regarded as the steam-generating portion.

The process in a drum-type boiler occurs in a different manner (Fig. 23.3). The mass filling of the boiler is several times greater than in the once-through type, and therefore, the steam flow rate D will for a certain time be higher than the flow rate of feed water G, leading to a lower water level h in the drum. On the other hand, since the pressure in the water-wall tube system and drum increases with D, some heat must be spent for heating the water contained in the drum to the corresponding boiling point. For this reason, D increases more slowly and for a certain time the proportion between B and D is disturbed and results in a higher t_{ss} . As D approaches a new static value, steam temperature ceases to rise and then drops down to the original or a new level which is determined by the static characteristic of the superheater (whose dimensions are not changed in the regime considered). It should be clear that the considered pattern of the process is true only for the period in which water level in the drum decreases to a lower allowable limit.

If the flow rate of food water in the once-through boiler decreases at constant B, the process will occur essentially in the same way as shown in Fig. 23.2. In the drum-type boiler in this case, the heat consumption for heating water in the drum decreases, resulting in a higher steam generation in the water walls and higher steam flow rate through the superheater. Since the dimensions and heating load of the superheater remain unchanged, the temperature of steam drops somewhat down. This process, as the previous one, occurs only in the period when the water level in the drum falls down to the lowest mark. A different process takes place if fuel combustion is increased suddenly. In the once-through boiler, this leads to 'outhurst', i.e. vigorous steam generation and carry-over of moisture by steam along the steam path. In this case, the length of the steamgenerating portion increases sharply (Fig. 23.4b), resulting in a shorter superheating portion and lower live steam temperature. As the additional steam passes from the boiler, the live steam temperature begins to rise and may exceed substantially the original level (Fig. 23.4c). The process that takes place in the drum-type boiler is similar, in a certain respect, to that described. The water level in the

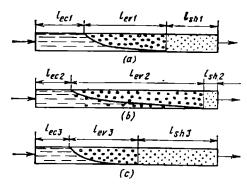


Fig. 23.4. Displacement of boundaries of the steam-generating portion in a once-through boiler on a sharp increase of fuel consumption (a) conditions before disturbance; (b) after disturbance; (c) final steady-state conditions

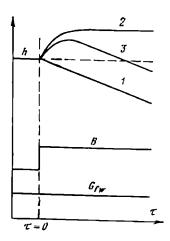


Fig. 23.5. Variations of the water level in the boiler drum on a sharp change of fuel consumption

drum 'swells' to a height as much as a few hundred millimetres. The disturbance of the material balance and the swelling of water in the circulation system have opposite effects on the water level in the drum (Fig. 23.5). An increase in D above G_{fw} results in a lower h (curve I), while swelling leads to a sharp rise in h which is then stabilized (curve 2). Using the superposition method, one can find the resulting change in h (curve 3). This curve shows that the effect of water swelling is predominant at the initial stage of the process and that of a change in the material balance, at the final stage. In contrast to the process in the once-through boiler, the drum in this case serves as a trap for entrained moisture, and therefore, t_{ss} changes much in the same manner as on a common increase of fuel consumption.

The effect of variation in the feed water temperature t_{fw} in the two boilers is also different. In the once-through boiler, a change of t_{fw} , under other conditions unchanged, results in the displacement of the boundaries of steam-generating portion in a similar manner as shown in Fig. 24.2, and in an increase of t_{ss} . In the drum-type boiler, the quantity of heat consumed

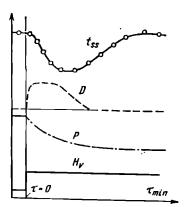


Fig. 23.6. Variations of the operating conditions of a boiler on opening of turbine control valves

for preheating water in the drum decreases, which leads to an increase of the steam flow rate through the superheater portion and a decrease of $t_{\pi\pi}$. If the flow rate of feed water increases and its temperature decreases, the processes occur in reverse direction. An exception is the process of 'outbursting' of moisture in the oncethrough boiler, which can only be observed on a sharp rise of heat release the furnace. With an external disturbance of the regime, the processes in the once-through and drum-type boilers are essentially similar. For instance, as the opening H_n of the control valves of the monobloc turbine increases, the pressure of live steam and the saturation temperature diminish. This results in a lower filledin mass of the boiler, higher steam generation, and lower temperature of superheated steam (Fig. 23.6). In the drum-type boiler, this process takes more time, since the mass filling should be changed more substantially. The change in t_{ss} in the once-through boiler is greater than in the drum type, since in addition to an increase in steam flow rate, the length of the steam-generating portion diminishes (the boundary of this region shifts downstream due to an increase of the enthalpy of dry steam on a decrease in pressure).

With t_{ss} maintained at the specified value, the process considered makes it possible to utilize the accumulating capacity of the boiler for quick variation of the turbogenerator load when it is needed to control the frequency and energy transfer in the system. The process has a definite effect on the unit operation in a regime of variable pressure of live steam. Load shedding in the unit is first carried out at a constant pressure of live steam by closing the control valves of the turbine. With further unloading of the unit, the position of the control valves remains unchanged and the pressure of live steam decreases. This regime offers certain advantages in boiler operation. In particular, the efficiency of the unit increases due to lower throttling of steam in the control valves of the turbine and smaller energy consumption by the feed pump. The range of the unit loads in which the temperature of reheated steam can be maintained at nearly nominal value becomes wider since, as the unit is unloaded, the temperature of steam at the inlet to the reheater is not decreased but increased. These advantages are mainly revealed at supercritical pressures. On the other hand, the variable-pressure regime decreases the rate at which the boiler load can be raised since part of heat is accumulated in the working fluid and tube metal. This drawback, however, can be decisive for the selection of the operating regime only in exceptional cases.

The combination of the processes discussed determines the dynamic properties of a boiler. It should be emphasized that the variable component of mass filling and the accumulating capacity of a boiler are decisive in this respect. Assuming these factors to be ineffective $[d\ (G_{in})/d\tau=0]$ and $d\ (Q_{in})/d\tau=0]$, it follows from equations (23.2) and (23.3) that the flow rate of steam should be equal at any moment to the flow rate of the feed water and that the temperature of superheated steam should follow the

variations in the heat release in the furnace without time lag. Under such conditions, a boiler would be uncontrollable. Under real conditions, the inertia of transient processes forms a certain time reserve essential for effecting the control process.

For a drum-type boiler, a change in load is done by changing simultaneously the fuel consumption and the capacity of blowers, with subsequent variation of the feed water flow rate which is controlled by the water level in the drum, with leading pulses from variations of the steam and feed water flow rates. The load of a once-through boiler can be varied by similar procedures, with the sole difference being that feed water flow rate is controlled by the fluid temperature in an intermediate portion of the path. A different control procedure is often employed in the automatic control systems of once-through boilers: first the feed water flow rate is varied according to the load and the flow rates of fuel and air are controlled in proportion to it and corrected by the fluid temperature in the intermediate portion of the path. In monobloc units, two versions of load variation are employed. For planned load variations, the load of the turbine is changed by varying the steam-generating capacity of the boiler so as to maintain a constant pressure of live steam. When the unit operates in a regime controlled by the power controller of the turbine, the load of the turbogenerator is changed first and the boiler load is then corrected. Methods to control the temperatures of live and reheated steam and the steam temperature in the boiler path have been discussed in Sec. 18.4.

23.4. Starting-up Circuits of Monobloc Units

In addition to the main elements of the thermal diagram of a boiler, the starting-up circuit includes special starting devices and pipelines which are used only at start-up, shut-down and lead shedding of the boiler. In conventional boilers, these include devices for discharging the steam from the boiler into the atmosphere before the boiler is connected to the steam main. Once-through boilers should in addition be provided with means for the drainage of water into a drainage tank during the period before steam superheating is started in the boiler. In monobloc units, the start-up and shut-down regimes of the boiler and turbine are interrelated more closely and are much more complicated. In this connection, starting-up circuits are specially developed for monobloc units at the design stage. In Soviet practice, starting-up circuits, operating instructions and automatic control systems for monobloc units have been typified. They are based on a standard procedure of starting up a unit by 'sliding' regime, i.e. by gradually increasing the flow rates, pressures and temperatures of live and reheated steam. This procedure creates the most favourable conditions for starting the turbine and decreases starting-up losses. The procedure requires no special devices for its realization in monobloc units with drumtype boilers. On the other hand, the drum has a large mass of metal and thus limits the rate of pressure rise in the boiler (and accordingly the temperature of saturation) during start-This limitation primarily holds for the initial period of boiler firing (to a pressure of 0.6 MPa) when the temporature of saturation increases most rapidly with increasing pressure. During this period, water retained in superheater coils may often prevent free passage of steam from the drum and the pressure in the latter rises more rapidly.

For the quick starting-up of a boiler, the starting circuit has a pipeline for steam discharge from the drum into the atmosphere (Fig. 23.7, I), which is in use during the period when prossure rises roughly to 0.6 MPa. The sliding regime of starting-up of a once-through boiler cannot be performed without special devices, since

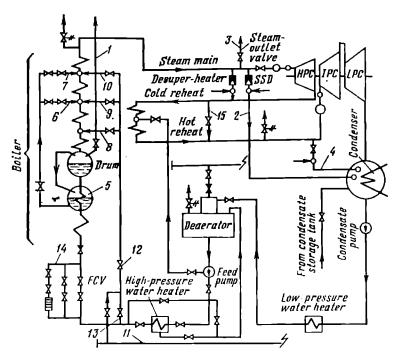


Fig. 23.7. Starting-up circuit of a 200-MW monobloc unit with a drum-type boiler Abbieviations in the diagram: FCV—feed-water control valve; SSD—startup-shutdown device; HPC—high-pressure cylinder; IPC—intermediate-pressure cylinder; LPC—low-pressure cylinder

from the requirements of reliable temperature conditions and hydrodynamics of steam-generating water walls, the starting-up flow rate of feed water must not be less than a specified value (usually 30% of the rated flow rate) and the fluid pressure in the water walls must be maintained at a level close to the nominal pressure. For a sliding starting-up regime, a oncethrough boiler is provided with an internal starting-up unit (Fig. 23.8) which comprises an internal gate valve, an internal separator, and pipelines with throttle valves for the supply of the working fluid to the internal separator (Thr1), drainage of moisture (Thr2), and steam removal from the internal separator (Thr3). The internal gate valve is arranged in an intermediate portion of the superheater path of the boiler. The rise in the enthalpy of the fluid in the path behind the internal gate valve at the rated load is usually 590-630 kJ/kg. The internal gate valve limits the throughput capacity of the internal separator and throttle valves. The internal separator ensures an increase in the steam-generating capacity of the boiler by a sliding regime with a constant (starting-up) feed water flow rate. The quantity of steam separated in the internal separator is determined by the moisture content of the st amwater mixture supplied, which is controlled to the specified value by varying the fuel consumption. During that period of starting-up, a once-through holler may be likened to a drum-type boiler. By means of the throttle valve Thr1, the pressure in the steam-generating water walls of the boiler is maintained close to the working pressure. During starting-up of the unit, the pressure of live steam (and therefore, the pressure in the internal separator) is determined by the steam-

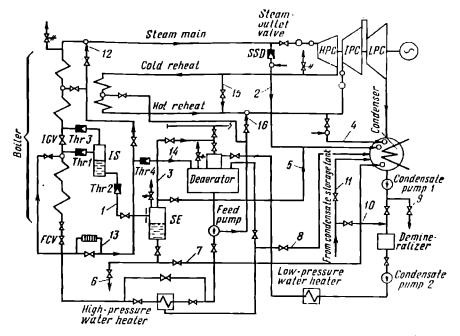


Fig. 23.8. Starting-up circuit of a 300-MW monobloc unit with a once-through boiler Abbreviations: SSD-startup-shutdown device; HPC-high-pressure cylinder; IPC-intermediate-pressure cylinder; LPC-low-pressure cylinder; IGV-internal gate valve; Thr-throttle valve; IS—internal separator; SE-start-up expander; FCV-feed-water control valve

generating capacity of the boiler and the positioning of the control valves of the turbine. The throttle valve Thr2 serves to regulate the removal of moisture separated in the internal separator. The throttle valve Thr3 is used mainly to cut off the steamsuperheating path behind the internal separator during the initial period of boiler starting when the effectiveness of the separator is still insufficient (at the steam content of the fluid less than 8-10%) and to control the steam flow rate through the superheater when it is connected to the circuit. The starting-up circuit for any type of boiler has a bypass line around the turbine (2 in Figs. 23.7 and 23.8) which connects the steam main directly with the condenser. A startupshutdown device in this line comprises a closing throttle valve, a noise absorber, and a spray-type steam attemperator. Live steam is discharged through the turbine bypass line in the following cases:

- (a) at starting-up of the unit, for a period when steam temperature is raised to the level determined by the thermal conditions at the turbine inlet;
- (b) at shut-down, for the period during which steam is discharged from the boiler and the latter is cooled; and
- (c) at load shedding, for the period when the turbine is idle-running or supplies only the auxiliary load of the unit.

In the USSR, single-stage bypassing of the turbine is the standard solution for monobloc units of 200-MW capacity and more. Since the steam discharged through the startup-shutdown device is not supplied to the reheater, the latter is arranged in a zone of reduced temperatures of gases. The boiler is allowed to operate with no steam supplied to the reheater at a fuel consumption up to 30% of the rated value.

In some monobloc units (150-MW, etc.), two-stage turbine bypassing is employed (Fig. 23.9). Live steam is

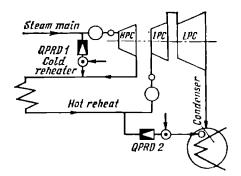


Fig. 23.9. Diagram of two-stage bypassing of turbine

Abbreviations: IIPC—high-pressure cylinder; IPC—intermediate-pressure cylinder; LPC—low-pressure cylinder; QPRD—quick-action pressure-reducing desuperheater

discharged through the first bypass line into the cold reheat pipeline and is discharged from the hot reheat line through the second bypass line into the condenser. In this circuit, the reheater can be located in the zone of elevated gas temperature, which widens the load range of the unit at the rated temperature of reheated steam. This, however, increases the use of expensive austenitic steel in the plant. Besides, for sudden shedding of the load, it is essential to employ pressure-reducing quick-action superheaters. The starting-up control of the unit is more complicated, since the steam flow rate through the quickaction desuperheater should be controlled according to the specified ratio of flow rates through the turbine cylinders. For these reasons, the twostage bypassing scheme is not popular in the USSR.

The start-up circuit for a drumtype boiler has, in addition, a pipeline (3 in Fig. 23.7) through which the steam accumulated in the boiler is discharged to the atmosphere when the unit should be shut down when vacuum is broken in the condenser.

In starting-up circuits, the heat of the steam discharged through the turbine bypass is not utilized, since this is not economically justified in view of the short time of the regime.

In some monobloc units, live steam can be supplied through a pressurereducing desuperheater into the reheater system to heat up the latter (see Fig. 23.7). After that, steam from the hot reheat pipeline before the intermediate-pressure cylinder is discharged through steam-discharging lines 4 into the condenser. In oncethrough boiler units, the circuit has means for utilization of the heat of fluid discharged from the internal separator. The fluid is fed through pipeline 1 (see Fig. 23.8) into the second separation stage — the start-up expander (SE); saturated steam from the expander is supplied into a deaerator where it gives up its heat to the feed water (line 3 in Fig. 23.8). Under certain conditions (for instance, at start-up of the unit after a short standstill), the quantity of steam separated in the expander may be higher than the quantity that can be supplied to the deaerator, and the surplus of steam is discharged through pipeline 5 into the condenser.

The steam removed from the expander entrains only a small fraction of the impurities present in the water drained from the internal separator (they are mostly dissolved in water). Thus, when this steam is fed from the expander to deaerator, it practically does not impair the quality of feed water in the boiler. Contaminated water can be discharged from the expander through a drainage pipeline into the circulation water conduit 6. When water in the expander is sufficiently clean, it is delivered through pipeline 7 into the condenser and then passes through the demineralizer.

In order to ensure the specified water conditions, the starting-up circuit of a monobloc unit with a once-through boiler has pipeline 8 through which water (contaminated on washing the steam space during start-up) is discharged from the high-pressure water heaters into the condenser. To remove dirty condensate from the circuit during the turbine washing, there is provided drainage pipeline 9 (see Fig. 22.8)

after the first-stage condensate pumps. During that period, clean condensate is supplied from the condensate storage tank through pipeline 10 immediately before the demineralizer. During the unit start-up, pipeline 11 supplies clean condensate to the condenser.

Starting-up circuits have special devices to maintain the temperature of live and reheated steam. A characteristic feature of drum-type boilers is that their steam-generating capacity during start-up lags substantially behind the increasing fuel consumption, so that the temperature of tubes of the superheater rises substantially and may exceed the allowable limit. Besides, it is essential to maintain the specified schedule of live-steam temperature variation before the turbine. On the other hand, the condensatespraying system of the main circuit (5-7 in Fig. 23.7) can operate only at boiler loads not less than 30% of the rated value. For this reason, there is provided a starting-up spray system with atomizing nozzles arranged in the condensate-spray diffusers. The first two groups of the spray nozzles (8, 9) are intended to hold the temperature of the platen superheater section tubes within the safe limits, while the third group (10) maintains the specified schedule of the livesteam temperature growth. This group can be arranged in the circuit before the outlet stage of the superheater if this has favourable dynamic properties and a limited rise of steam enthalpy (up to 170 kJ/kg at the rated load). In once-through boiler units, this condition cannot usually be observed, so that the starting spray group is provided in the steam main (12 in Fig. 23.8). Moreover, the heataccumulating capacity of once-through boilers is relatively low and there is no need to employ spraying to protect the superheater stages against overheating.

In the sliding starting-up regime, the pressure of feed water is much higher than that of live steam. The pressure of water in a once-through boiler unit, which is continuously fed with water, is reduced by means of a constant-flow rate circuit comprising a bypass line 13 (see Fig. 23.8) with a set of orifices and a line 14 with throttle valve Thr4 for recirculating water into the deaerator. The valve Thr4 maintains a constant pressure 'before itself', which makes it possible to redistribute water between the starting spray and deaerator at a constant total water flow rate and in the set of orifices at a constant pressure gradient. This circuit allows water to be supplied to the starting spray from the discharge of the feed pump.

This scheme is inapplicable in circuits with drum-type boilers, since a boiler is fed with water during a certain period of firing from neighbouring units through a crossover feed water pipeline (11 in Fig. 23.7), so that recirculation of water to the deserator of the unit being started would cause overfilling. Because of this, the pipeline for supplying water from the crossover line 11 is provided with the main control valve 12 in which the excessive pressure gradient is reduced. When the boiler is changed to feeding from its own feed pump, the starting sprays are switched over to the feed water pipeline *13*.

The foed water supply to the drumtype boiler is controlled by means of bypass lines 14 with a reduced throughput capacity and the main feed water control valve. In the once-through boiler circuit, only the main feed water control valve is provided, since the minimal flow rate of feed water is much higher than in the drum-type boiler.

In most cases, the starting regime of a monobloc unit should ensure a reduced reheated steam tomperature level and further control of this temperature according to the specified schedule. In view of the dynamic properties of the reheater and some other factors, the main means of control can be used only beginning from loads of 25-30% of the rated value. A method for starting control that is widely

used in all types of monobloc units is steam bypassing in the reheating system (line 15 in Figs. 23.7 and 23.8). Part of the steam is passed from the cold reheat line into the hot reheat line, thus lowering the temperature gradient, and therefore, the heat absorption in the reheater. As steam from the cold reheat line is mixed with that in the hot reheat line, the temperature of the reheated steam can be controlled at a lower level. In oncethrough boiler units, a starting spray is additionally provided in the steam main (16 in Fig. 23.8), which is used to control the temperature of reheated steam upon closing the steam bypass. Feed water to the starting spray is supplied at a pressure of more than 7 MPa from the intermediate stage of the feed pump, so that there is no need to control the water pressure in the spray line. In some cases, only one means of reheated steam temperature control is employed. It is possible to employ steam bypassing only, if it has been established by boiler tests that the reheater has reliable temperature conditions at the maximum load at which the steam bypass will be used. The allowable conditions for starting spraying are determined by the minimal flow rate and pressure of steam at which moisture will be evaporated completely in the steam flow and will not precipitate on the internal surface of the steam main, especially in the nearest bend after the spray. Depending on the test results, the distance from the spray to the nearest bend of the steam main should be not less than 18-25 m.

The starting circuits discussed above have been employed in monobloc units of a capacity of 500, 800 and 1200 MW. Their specifics are determined by the application of feed pumps and in some units, of air blowers driven by steam turbines. To this end, the circuits can be supplied by steam from other plants to drive these turbines and also can be fed with live steam (through the startup-shutdown device of the quick-action desuperheater) at

load shedding. Starting-up circuits of monobloc units are described in more detail in [12, 43].

23.5. Shut-down and Load-shedding Regimes

Before normal (not emergency) shutdown, the boiler of a monobloc unit should be unloaded, i.e. its load should be shedded. When shutting down for a short time (for instance, overnight), the hot state of the plant is usually retained wherever possible, so that load shedding is limited (for the most part not less than 50% of the rated load). When shutting down for a longer time, it is advisable to utilize the accumulating capacity of the boiler as much as possible. This is favoured by the deep unloading of the unit with sliding (variable) pressure of live steam. In addition to this, upon flame extinction in the boiler furnace, the turbogenerator is left for a certain time connected to the system so as to work off the accumulated steam in the boiler. The load-shedding conditions in a drum-type boiler are determined by the rate at which the saturation temperature in the drum decreases; this should not exceed 1.5-2 deg C/min. The rate at which load shedding occurred in a once-through boiler by the action of its automatic control system is determined only by the dynamic properties of the boiler.

A boiler is shut down by interrupting the fuel supply to the burners and the supply of feed water. In drumtype boilers, the drum is first filled with water to the upper working level. Upon flame extinction in the furnace, the draft fans remain in operation for 10-15 minutes to prevent the accumulation of an explosive mixture in the gas-air path, which is thus ventilated. During the standstill, measures are taken to prevent the condensation of steam in superheater tubes, since this might involve complications in subsequent firing of the boiler and decrease the boiler reliability. In view of this, as a once-through boiler is

shut down to reserve and the feed water supply to it is discontinued, the water path up to the internal gate valve is closed (by closing the feed water valve, internal gate valve, and the fittings of the internal separator) and steam is discharged from the superheater (via the startup-shutdown device) and reheater (via the discharge line from the steam mains) into the condenser. When shutting down a drum-type boiler, the pressure in it is reduced gradually (by steam discharging through the startup-shutdown device of the desuperheater), so as to maintain the saturation temperature below the temperature of flue gases in the superheater, i.e. by forming conditions preventing the condensation of live steam in the superheater. The drum is periodically fed with water. The reheater is freed from steam in the same way as in a oncethrough boiler.

When the cold shut-down of a monobloc unit must be performed, requiring that all boiler equipment be cooled, load shedding is carried with the gradual reduction of the pressure of live steam. In a once-through boiler, the internal gate valve is closed at a load roughly 60% of the rated value and the pressure in the path up to this valve is then maintained constant. In both types of boilers, the temperature of live steam is lowered together with its pressure at a rate allowed by thermal stresses in massive metallic elements of the boiler, pipelines and turbine. At the same time, the temperature of reheated steam is lowered. The superheater and reheater can thus be cooled roughly to 300°C.

When it is needed to cool a drumtype boiler and steam pipelines for repairs, as the boiler furnace is extinguished and the turbine switched off, the steam accumulated in the boiler is discharged into the steam main of the station, the turbine condenser or the atmosphere. The water level in the drum is raised to the upper mark (roughly 200 mm below the top surface of the drum); this measure can minim-

ize the temperature difference between the top and bottom of the drum (which should be not more than 40 deg C). At a pressure of 0.6-0.8 MPa in the boiler, the gas-air path is intensively ventilated. Once-through boilers have no massive metal elements (such as the drum) and can be cooled more quickly. The load shedding of a unit up to the furnace extinction and turbine switch-off is carried out at the rated pressure of live steam. In supercritical-pressure boilers, steam discharging is then started through the steam drainage before the steam-outlet valve, with the startup-shutdown device being closed. When the temperature of live steam has dropped down roughly to 420°C, the feed water supply to the boiler is restarted at a flow rate of 10-15% of the nominal value. Due to the heat accumulated in the boiler, the temperature of fluid along the path decreases gradually. As the water temperature drops down roughly to 300°C, the flow rate of the feed water is increased up to 30% of the nominal value and water is pumped through the path until the complete cooling of the boiler and pipelines is accomplished. In subcritical-pressure monobloc units, the once-through boiler and pipelines are cooled by gradually decreasing the saturation temperature at the superheater outlet. To perform this, as the feed water supply to the boiler is restored at 10-15% of the nominal flow rate, the pressure in the boiler is maintained constant (by means of the startup-shutdown device) until the temperature of fluid at the boiler outlet drops down to the saturation point. The feed water flow rate is then increased to 30-40% of the nominal value and, since the boiler has accumulated an appreciable quantity of heat, the fluid at its outlet is for an appreciable time in the state of a steam-water mixture. The reduction rate of the saturation temperature can be controlled properly by regulating the pressure reduction rate of the fluid; this ensures the desired cooling rate of the plant. Similar methods are employed for cooling the path of oncethrough boilers up to the internal gate valve.

In emergency situations, a boiler is shut down with 'pressure retention'. As the furnace is extinguished, the feed water control valve and steamoutlet valve are closed, the turbogenerator is switched off, and the startup-shutdown device is kept closed if the pressure of the live steam is below the actuating pressure of the boiler safety valves. In this method, the pressure and temperature of fluid along the boiler path are first hold at the same values as in normal operation. After that, they diminish slowly owing to heat transfer to the surroundings and loose fittings of the watersteam path. If the trouble is climinated quickly, this method of shuttingdown makes it possible to start the boiler from the state of hot reserve. Otherwise, shutting-down is continued and steam from the superheater is discharged.

Emergency shut-down of a boiler must be carried out immediately in the following situations:

- (1) when the water level in the drum passes beyond the safety marks or the supply of feed water to one of the flows in a once-through boiler is interrupted for more than 30 s;
- (2) on failure of the water-level gauges in a drum-type boiler or of the feed water flowmeters in a once-through boiler;
- (3) when there is no flow of steam through the reheater;
- (4) when the pressure has risen intolerably (in once-through boilers, a pressure drop is also dangerous);
- (5) on rupture of tubes in the watersteam path;
- (6) on flame extinction in the furnace, explosions in the furnace or gas ducts, or inflammation of combustible deposits in convective heating surfaces:
- (7) on an intolerable reduction of the pressure of gas or fuel oil behind the control valve;
 - (8) when there is no voltage at re-

mote-control devices and measuring and control instruments; and

(9) on switching off of the turbine (in monobloc units) or of some of the auxiliary equipment (draft fans, blowers, etc.).

In all such cases, a delayed shutdown can aggravate the situation and cause serious damage, so that the boiler operator is instructed to stop the boiler without waiting for permission from the station management or load dispatcher of the power system.

An emergency shut-down of a boiler is effected by the protection system on receiving the signal on the appearance of some or other emergency situation. If the protection system fails, the required procedures should be carried out by the operating personnel. These functions are facilitated by provision of light and audio signalling means on the control board of the unit.

In certain situations, a boiler can be shut down only upon agreement with the station management, since the development of a failure and serious consequences are only probable after a cortain time. Such situations include the appearance of small failures (cracks) in the water-steam path elements, a rise in the temperature of the metal of particular heating surfaces. low quality feed water, failures in the protection systems, automatic controllers, measuring and control instrumonts, remote-control devices, etc. In such cases, measures first should be taken to eliminate the trouble and if these fail, it should be decided how long the boiler can be in operation before shut-down.

If a group of the auxiliary equipment is switched off, the boiler load should be immediately shedded to avoid heavy damage to the equipment. For instance, if one of the two induceddraft fans has been stopped, the flame will shoot out from viewholes, blowdown holes, etc. and can cause fire in the boiler room, damage to the boiler structure, etc. If one of two forceddraft fans is stopped, this may cause

flame extinction, the accumulation of unburned fuel in the furnace and its explosion which can lead to serious consequences. As may be clear from these examples, the prime task in such situations is to diminish the fuel consumption to a sufficient level for operation of the remaining equipment. At the same time, it is essential to control the flow rate of feed water and the temperatures of live and reheated steam and to change the load of the steam consumer and of the auxiliary equipment that still remains in operation. In some cases, certain switchings should be made in the circuits of the water-steam and gas-air paths. All these procedures are performed by the protection system or special systems for automatic load shodding of the unit. Nonetheless, the operating personnel should be always ready to perform these procedures by means of remote-control devices. For this reason, the personnel is periodically trained to cope with emergency situations. Training simulators are used effectively for the purpose.

An extreme case among the above is load shedding of a unit to a level at which the boiler supplies only its auxili-

ary equipment.

As has been demonstrated by boiler tests, a drum-type boiler in such cases can be shut down while the turbogenerator continues operation by consuming the steam accumulated in the boiler. In monobloc units with oncethrough boilers, the load of the boiler in such a situation decreases rapidly to the firing load (roughly 30% of the rated load). In boilers fired on solid fuel, part of the burners are closed (together with their fuel-supplying means) and fuel-oil burners are switched on in order to 'brighten' the flame. In some cases, the furnace should be changed completely to fueloil firing. At the same time, all the procedures required for partial load shedding are performed, but since the regime has been disturbed more substantially, it is more difficult to stabilize it. If these procedures are not

properly conducted, this can lead to emergency situations. In boilers with a single bypass in the starting-up circuit, it is especially important to cut down the fuel consumption as quickly as possible, since otherwise 'burnthrough' of the reheater tubes is likely to occur. In view of this, it is prescribed by boiler rules that a monobloc unit can continue its operation only if its protection system and automatic control systems are intact.

23.6. Regimes of Boiler Firing and Unit Starting

These regimes include the following three main stages: proparatory procedures, boiler firing proper, and raising the load to the specified value. These regimes will be described as applied to monobloc units, the most advanced equipment of modern thermal power stations. The first, 'preparatory', stage includes 'assembling' the water-steam, fuel, and gas-air paths, preparing all mechanisms and systems, creating vacuum in the turbine condenser, pre-starting descration of feed water, etc. A drum-type boiler is filled with water as required. The water level in the drum should be somewhat below the normal mark so as to allow 'swelling'. A once-through boiler is to be filled with water in all firing regimes, except for firing from the state of hot reserve. As water is fed into the boiler, it displaces air from the system (provided that the air pressure is not excessive). In a once-through boiler, the feed water flow rate is adjusted at the starting-up level and the water pressure is raised to the working value by closing the throttle valve Thr1 (with the internal gate valve being closed). In boiler firing from the hot state, first a reduced flow rate of water is established (10-15% of the nominal value), which makes it possible to slowly cool the boiler path up to the internal gate valve and internal separator. The starting-up flow rateTof water is set in upon raising the water pressure

before the internal gate valve. Water from the internal separator is discharged into the start-up expander SE and further into the circulation conduit (6 in Fig. 23.8). The startupshutdown device is opened to create vacuum in the superheater (except for cases of starting from hot reserve). This procedure is also carried out in a drum-type boiler if there is no excessive pressure in it; this ensures a slower rise in the saturation temperature of the drum during firing. In cases when the startup-shutdown device is initially closed, its opening is performed only upon firing the furnace, so as to maintain a constant pressure of the live steam that has been generated to that moment.

During boiler standstills, moisture may accumulate in some stages of the superheater, though certain measures, as indicated in Sec. 23.5, have been taken to prevent this. Besides, untightness of the internal gate valve and throttle valve Thr3 in a oncethrough boiler may lead to moisture accumulation in the pipeline and the first heating surface behind the internal gate valve. This moisture can be 'pushed out' into the hot headers of the boiler during firing and cause their cracking. In a drum-type boiler, this may result in a quicker pressure rise in the drum at the initial stage of firing, which, in turn, diminishes the allowable heating intensity in the furnace. By opening the startup-shutdown device, the superheater is connected with the condenser which promotes the evaporation of moisture from the superheater tubes during boiler firing.

After switching on the draft fans, ventilating the gas-air path, and preparing fuel-supply mechanisms, the regime of burner firing is carried out (firing fuel-oil or gas burners are ignited). It is advisable to carry out the boiler firing by igniting as many burners as possible with the least fuel flow rate through each of them; this is essential for the uniform heating of the water walls around the furnace

periphery and for minimizing local thermal stresses; in drum-type boilers, this measure is additionally essential for the development of circulation in all the water walls. As established by tests of the existing drum-type and once-through boilers in the USSR, the flow rate of fuel during the initial period of firing should not exceed 20% of the nominal value. Under such conditions, the tube wall temperature of superheating surfaces does not exceed the allowable limit even in noload operation of the boiler. In boiler starting from a cold or warm state, the initial flow rate of the fuel is controlled at a level of 12-15 % of the nominal value. In a drum-type boiler, this ensures the quick development of circulation in water walls and, on the other hand, the rate of pressure rise in the drum is within the tolerable limits (with steam discharge from the drum to the atmosphere or with the drainable superheater). Irrespective of the type of boiler, the indicated flow rate of fuel is sufficient for heating the steam pipelines.

In starting from the hot state, the fuel flow rate in the initial period is established at 20% of the nominal value or, if there is steam flow through the superheater, is increased additionally so as to attain the required temperature of live and reheated steam.

As the initial flow rate of fuel has been established in a once-through boiler, the flow rate of the feed water and the pressure of fluid before the internal gate valve are maintained constant. As the pressure of fluid in the start-up expander SE has risen up to 0.4-0.5 MPa, steam from the expander is removed to the deacrator and, when a sufficient quantity of drain water has been accumulated, the cycle is closed (by switching over water discharge from the start-up expander from the circulation pump to the condenser). A drum-type boiler during this period of firing is periodically fed with water from neighbouring monobloc units (lines 11 and 13 in Fig. 23.7) so as to maintain the water

level within tolerable limits. In boilers with boiling-type economizers, the regime with periodic water feeding and a low constant flow rate of water can in some cases lead to a substantial maldistribution of heat and hydraulic maldistribution. Some water-distributing tubes may supply the fluid of a higher enthalpy (up to superheated steam) to the drum. This is prevented by maintaining the specified temperature of fluid in an intermediate section and at the economizer outlet by properly controlling the flow rate of water and by increasing the rote of blowdown on an increase in the water level in the drum.

As the initial flow rate of fuel has been established, it is possible to raise the flow rate and parameters of live steam in a drum-type boiler or the temperature of fluid before the internal gate valve (t'_{IGV}) , in a oncethrough boiler. The temperature of the fluid determines its moisture content on entry to the internal separator. As found from tests, the internal separator can operate efficiently at a dryness fraction of 8-10% (t'_{IGV} = $-250\text{-}270^{\circ}\mathrm{C}$), and therefore, it is possible to connect the superheater to the circuit. This procedure is performed by gradually opening the throttle valve Thr3 (in steps of 10-15% in 2-3 minute intervals). As the superheater is connected to the circuit, the temperature of tube metal in the heating zone decreases. At the same time, the temperature of steam at the boiler outlet gradually increases, since the heat-transfer coefficient az increases with the flow rate of the steam. Since the throttle valve Thr2 is still fully open at this stage of starting-up, part of the steam is discharged together with water from the internal separator into the start-up expander. Thus, the next procedure is to close the valve Thr2 which is carried out to ensure removal of all moisture and part of steam (roughly 5% of the flow rate of drain water) from the internal separator, which increases its efficiency. Further, as the dryness

fraction of the fluid in the internal separator increases, the valve Thr2 is closed more, until it is completely closed upon appearance of superheated steam before the internal separator, which is an indication of a change from a separating regime to a oncethrough regime having taken place in the boiler.

As the steam flow rate through the superheater increases, the steam mains are gradually heated. Steam is discharged from them through the startup-shutdown device and the drainage of the dead-end sections. Heating is usually carried out until the temperature of steam before the highpressure cylinder of the turbine is roughly 100 deg C higher than the temperature at the steam inlet. In units provided with a pressure-reducing desuperheater (see Fig. 23.7), the reheater system is heated by supplying live steam to the cold reheat pipeline and discharging it from the hot rcheat pipeline into the condenser. This heating is only started when the temperature of steam before the desuperheater exceeds the temperature at the discharge end of the high-pressure cylinder of the turbine, so as to avoid cooling of that section. The end of heating of the hot reheat pipeline is determined by the temperature of steam before the intermediate-pressure cylinder of the turbine which exceeds the temperature at the steam inlet by 50-80 deg C. In supercriticalpressure monobloc units for 300 MW and 500 MW, there is no desuperheater in the starting-up circuits (Fig. 23.8) and the rehenter system is heated by the 'combined' method. The turbine rotor is pushed by opening the control valves and the frequency of its rotation is then raised to 800-1000 rpm. Live steam through the high-pressure cylinder and the reheater system and is discharged from the hot reheat pipeline into the condenser, with the valves of the intermediate-pressure cylinder closed. As proven by tests, the rotors of the intermediate- and high-pressure

cylinders can operate at a low frequency without steam being passed through them. On the other hand, since only the high-pressure cylinder is actually in operation, the flow rate of steam is quite high and ensures quick heating of the reheater system. In some cases, vacuum in the turbine condenser is lowered in order to further increase the flow rate of steam through the system.

In some regimes, a monobloc unit can be started without preheating the steam pipelines, in particular, the regime of starting from the hot state. Moreover, when the thermal insulation is in proper condition, a unit can be started without preheating the reheater system after a standstill for 1-2 days. A criterion of such regimes is a decrease in the steam temperature by not more than 20-30 deg C compared with the temperature at the steam inlet to the turbine.

As preheating is completed, the flow rate of fuel is corrected so as to ensure the steam-generating capacity of the boiler at which the initial load of the turbo-generator will be around 5% of the rated value. In starting-up from a cold or a warm state, the flow rate of the fuel is usually established at the minimal level so as to obtain more easily the required low temperature of live and reheated steam. In starting-up from the hot state, however, the flow rate of fuel is increased to the upper limit (30% of the nominal value in the circuits with single bypass) to obtain a steam temperature close to the nominal value.

Before pushing the turbine rotor, the starting sprays are switched on and the temperature of live steam is established at a desired level. In a once-through boiler, the throttle valve Thr4 in the line of water recirculation to the deacrator is set so as to obtain a steam pressure before the starting spray nozzles that is 1.5-2.0 MPa higher than the pressure of live steam. In a drum-type boiler, the desired steam temperatures behind the stages of the superheater are additionally

preset. In some 200-MW and 300-MW monobloc units, the temperature of the reheated steam is controlled by the steam bypass. In units of higher capacities (500, 800 and 1 200 MW), there is no steam bypass and only the starting sprays to the steam main, which are put in operation before connecting the turbo-generator to the electric network, are used. Before connecting the turbo-generator to the network, when its rotor is accelerated and synchronized, the steam-generating capacity of the boiler and the temperature of live steam are maintained constant. During this period, the temperature of the reheated steam gradually increases for the same reasons as on connection of the superheater.

The temperature of the reheated steam rises especially sharply on connection of the turbo-generator, i.e. when the steam flow rate through the reheater system is almost doubled. For this reason, it is important to switch on in due time the devices which control the temperature of reheat. In units with once-through boilers, the startup-shutdown device is not closed during the period of turbogenerator synchronization, that all the control valves of the turbine are opened (and heated) due to a drop of pressure in the live steam. In units with drum-type boilers, the startup-shutdown device is partially closed to maintain a constant pressure of live steam, which improves the operating conditions of the boiler drum and control over the steam temperature. As the turbo-generator is connected to the network, the startup-shutdown device is closed and the unit begins to supply the initial

At the third stage of starting (loading) a monobloc unit, its elements are heated from the initial temperature to the final temperature corresponding to its operation under the rated conditions. An attempt to shorten the time of loading may result in the quick heating of the boiler elements

and the appearance of high temperature gradients. For instance, if a wall of a thickness h is heated at a rate V, $^{\circ}$ C/min, the temperature gradient across the wall is:

$$\Delta t = \frac{h^2}{2a} V \tag{23.5}$$

where a is the thermal diffusivity of the wall metal, m^2/h .

If a wall is heated at a constant rate V, the temperature stresses in it, $\sigma_{\Delta I}$, are linearly related to the temperature gradient:

$$\sigma_{\Delta t} = A\alpha E \Delta t$$
 (23.6)

where α is the coefficient of linear expansion, E is the elastic modulus of the metal, and A is a proportionality factor.

It then follows that the highest temperature gradients and highest temperature stresses can appear in massive thick-walled elements, such as the housing and rotor of a turbine, the drum and headers of a boiler, and fittings of the steam mains. In such cases, compressive stresses appear in the heated surface of an element and tensile stresses in its unheated surface. Upon heating an element, temperature stresses diminish to zero and often even change sign. Stresses of an inverse sign can form in elements on the reduction of the steam temperature or the shut-down of the unit. With frequently repeated startup-shutdown regimes, thermal stresses vary cyclically and can cause fatigue cracking of the metal. The number of cycles N to cracking depends on many factors, the principal one being the amplitude of stresses in a cycle: $\Delta \sigma = \sigma_{\max} - \sigma_{\min}$. The number of cycles N is inversely proportional to the square of $\Delta \sigma$. The allowable stresses in elements of a monobloc unit depend on the expected number of startup-shutdown regimes during its campaign. These stresses, in turn, determine the allowable heating rates of these elements.

In view of the above, the load of a unit should be increased strictly in accordance with the specified rate of raising the parameters of live and rcheated steam. As an example, Fig. 23.10 gives the starting curve for a 300-MW monobloc unit after a standstill for 60-90 h. As may be seen, the rate of raising the temperatures of live (t_{ss}) and reheated steam (t_{rh}) should differ depending on the initial thermal state of the turbine $(t_{\rm HPC},\ t_{\rm IPC})$, so as to ensure the most roliable regime of load application to the turbine. Similar conditions should naturally be ensured in monobloc units with drum-type boilers. Up to a load of 25-30% of the nominal value, steam temperature is controlled only by the starting devices. After that, the main control devices are switched on, and the starting devices are either switched off or used for the fine correction of steam temperature.

The pressure of live steam is increased in a 'sliding' mode. The way this

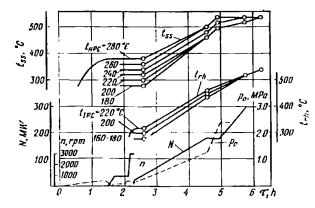


Fig. 23.10. Main characteristics of cold starting-up of a 300-MW monobloc unit

n-turbine rotor frequency; N-turbogenerator load; P_0- live steam pressure

mode is realized, however, depends on the particular characteristics of the equipment. For instance, in monobloc units with drum-type boilers provided with radiant superheater sections on furnace walls and a boiling-type economizer (such as the TGM-94 boiler), the pressure of live steam is raised by an accelerated schedule. Upon connecting the turbogenerator to the network, its control valves are set into a position in which the pressure of live steam rises to the nominal value, already at a load that is 40-50% of the rated value. In such a case, heat consumption for accumulation in the fluid and tube metal takes place at a reduced temperature level of the fluid, so that the load of the boiler can be raised more quickly with the temperature of the tube metal in the radiant superheater remaining within the tolerable limits. Besides, an increase in the pressure at low loads improves the thermal and hydraulic characteristics of the boiling economizer. A similar regime is employed in monobloc units with supercriticalpressure once-through boilers, the only difference being that the rated pressure of live steam is attained at a load roughly 60% of the nominal value and is determined by the throughput capacity of the starting unit of the boiler. As the load increases to this level and the steam pressure remains at the nominal value, the internal gate valve is opened. In 200-MW monobloc units with drum-type and once-through boilers, as the turbogenerator is connected to the network, the control valves of the turbine are opened fully and the rated pressure of live steam is attained only at the nominal load. In once-through boiler monobloc units, however, the throughput capacity of the internal separator and its fittings is not more than 60% of the nominal load. For this reason. as the nominal load is attained, the pressure of live steam before the turbine is raised to the nominal value and, at the same time, its temperature is increased so as to obtain a constant temperature behind the control valves of the turbine. The internal gate valve is then closed and the boiler transferred to the nominal pressure.

Solid fuel-fired boilers are transferred from starting fuel (gas or fuel oil) to solid fuel, if the load is 15-30% of the nominal value, and the consumption of the firing fuel is gradually reduced. As the boiler begins to supply the specified load, the elements of the starting circuit which are used only at start-up and shut-down are switched off and the voltage supply to the corresponding drive means is discontinued.

Conventional boilers (not in monobloc units) are started essentially in the same way, except for the specific procedures typical of monoblocs.

A special case is starting a oncethrough boiler from the state of hot reserve. For supercritical-pressure boilers, this regime is allowed provided that the pressure of the live steam has remained above the critical value after the period of standstill. In subcritical-pressure boilers, subcooling of the water at the injet to the lower radiation section should not be less than 15 deg C. Otherwise, as follows from experience, the water walls of the lower radiation section may incur substantial damage during boiler starting from the non-uniform distribution of fluid between the tubes (as regards its flow rate and enthalpy). With these conditions observed, the boiler is started by the method of quick transfer to the regime of normal operation. Since the parameters of the fluid vary only slightly during the period of standstill in hot reserve, starting is performed by controlling the starting flow rate of the feed wator, after which the fuel-oil burners are put in operation for 2-3 min, with the flow rate of fuel oil controlled according to the flow rate of the feed water. Since the flow rate of fuel lags somewhat behind that of feed water, the temperature of live steam first decreases (by 30-50 deg C), and is then restored to the rated level. The pressure of live steam is maintained constant by opening the startup-shutdown device. When these procedures are properly carried out, the starting of a boiler takes only 15-20 minutes.

Some monohloc units, especially those which supply a variable portion of the load curve, are started by an automatic process control system. In modern plants, such systems can perform automatic control of a given process and carry out discrete operations by means of logic control units. These units can switch on and off the mechanisms of the auxiliary system of a boiler, change the position (open or close) of closing valves, switch on and off automatic controllers, switch over the controllers from some actuators to others, after the structural schemes of the controllers, etc. Before performing a particular operation, a logic control unit checks that the operation is allowed. In monobloc units provided with an automatic control system of this type, the hoiler operator has the following additional functions:

- (1) to perform preparatory operations for starting the unit and select the reserve mechanisms which will be switched on automatically;
- (2) to supervise the functioning of the equipment and perform manual control should a failure occur in the operation of a particular automatic controller:
- (3) to correct the operating conditions (when needed) by adjusting the set-point devices of automatic controllers;
- (4) to check the state of the equipment upon completion of a particular stage of the starting regime and issue commands for a next stage.

Thus, the automatic control system of a monobloc unit is a complex of the technical means of control and the operating personnel which co-operates with these means.



STEAM GENERATORS OF NUCLEAR POWER STATIONS

24.1. Classification and Characteristics of Steam Generators for Nuclear Power Stations

According to the thermal diagram of a nuclear power station, steam is generated either directly in a boiling-type nuclear reactor or in heat-exchange steam generators in which heat is transferred from a coolant supplied from the reactor to the working fluid (water, from which steam is produced). Thus, the steam generator is an essential element of two- and three-circuit nuclear power stations. In single-circuit nuclear power stat-

ions, the function of steam generator is performed by the nuclear reactor proper.

Water, gases and liquid metals can be used as reactor coolants, and therefore, as the heat-transfer agents in the steam generator. An aqueous coolant in the primary circuit, as a substance having a low boiling point, cannot be heated to a substantially high temperature. This requires maintaining a high pressure in the primary circuit, which makes the whole path too complicated and expensive.

Liquid metals and gases have no limitations as regards their temper-

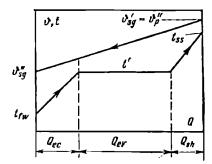


Fig. 24.1. t, Q-diagram of a steam generator with a gaseous coolant (superheated steam of a particular pressure)

ature at the reactor outlet. Liquid metals possess favourable thermophysical properties and can be easily heated to high temperatures without taking special measures to intensify heat transfer. On the contrary, gases have poor thermophysical properties and at atmospheric pressure cannot be used as coolants which could ensure suitable heat-transfer coefficients. It is possible to intensify beat transfer from gases by increasing the mass flow rate of a gas flow, i.e. by raising the gas pressure in a circuit. Therefore, steam of high or supercritical parameters can only be produced by using a gas or liquid metal, which are hightemperature coolants. For instance, if the coolant at the reactor outlet (or what is the same, at the inlet to the steam generator) has a temperature $\vartheta_s' = 600-650$ °C, it can ensure steam superheating to the standard parameters (p = 13-24 MPa and $t_{ss} = 545$ °C). Figure 24.1 shows the t, Q-diagram of such a steam generator. In this diagram, the temperatures of the coolant and working fluid are laid off as ordinates and the quantities of heat transferred to the heating surfaces of steam generator, as abscissae.

Water is a low-temperature coolant. If it is needed to generate steam at a pressure as high as possible (for instance, p=6.4 MPa and temperature of aqueous coolant at the inlet, $t'_1=320$ °C), the degree of superheating will be only 20 deg C. This is

illustrated in Fig. 24.2. Assuming the temperature gradient between the coolant (pressurized water) and working fluid to be $\Delta t = 25 \deg C$ so as to ensure active heat transfer, the degree of superheating that can be obtained at p = 6.4 MPa will be only $\Delta t' = 17 \deg C$. For this reason. nuclear power stations with aqueous coolant usually operate on saturated steam of a pressure of 5-7 MPa. When supplied to the turbine, wet steam $(\omega \leq 0.2\%)$ can cause erosion of steamdistributing devices and lower the turbine reliability. To prevent this effect, it is essential to superheat the steam slightly (usually by 20 deg C) so as to avoid moisture formation in steam-distributing devices.

As steam moves through the turbine, its moisture content increases. so that some stages of the turbine operate in the region of wet steam. which decreases the efficiency of the power station and causes erosion of the elements in the low-pressure cylinder of the turbine. In order toremove moisture, steam between the turbing cylinders is dried and superheated. Both processes are carried out in a separating superheater (Fig. 24.3). Steam from the high-pressure cylinder of the turbine is introduced tangentially at the top of the apparatus, passes through the separator, sweeps the superheating surfaces and is removed at a temperature of 241°C through the top to the low-pressure cylinder of the turbine.

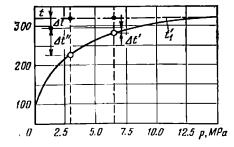


Fig. 24.2. Selection of the working fluid parameters for steam generators of nuclear power stations with water-cooled water-moderated reactors

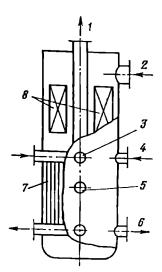


Fig. 24.3. Separating superheater t—superheated steam to low-pressure cylinder; the steam from high-pressure cylinder; the first-stage heating steam; the second-stage heating steam; the second stage; the surfaces; the separator

Steam generators of nuclear power stations may be of either vertical or horizontal arrangement. Those with a gaseous or liquid-metal coolant are usually of the vertical type. Steam generators of water-cooled water-moderated reactor stations may be of either vertical or horizontal type, each of which has its own advantages and drawbacks. A horizontal steam generator is simpler in manufacture and more reliable in operation. Vertical steam generators are more compact, but their elements are more intricate. With U-shaped coils, tube plates must be of a substantial thickness, so that fastening of tube ends requires a more expensive and complicated technology. The total cost of both types of apparatus is roughly the same.

A horizontal steam generator occupies several times the area of the vertical type of the same steam-generating capacity. A higher compactness of vertical-type steam generators is advantageous in view of radiation safety.

The performance characteristics and reliability of vertical and horizontal

steam generators are roughly equivalent. Both types are assembled and tested at the manufacturing works and can be transported by railroads. The dimensions of horizontal steam generators for a capacity of more than 250-300 MW are excessively large. The vertical design makes it possible to substantially increase the unit capacity; in this apparatus, the total cross section in the bottom portion is occupied by the heating elements and the separator is arranged in the top portion. For more convenient transportation, the heat-exchanger and separator sections are made separately and the apparatus is assembled on the construction site.

24.2. Steam Generators with Aqueous Coolant

There are two main types of steam generator for nuclear power stations with water-cooled water-moderated reactors, as regards the processes occurring in the secondary circuit. In the first type, the working fluid (water) boils in the bulk on the surface of submerged heating elements (naturalcirculation steam generators). In the second type, water moves in a tube system or in the intertubular space of a dense tube bundle (once-through steam generators). In both types, the heating surface is formed by a tube system in which the tubes may be straight, U-shaped, helical or assembled into plane curtains. The steamseparating devices of submerged-type steam generators are arranged either inside the generator drum (internal separators) or in a separate drum (external separators).

The principal types are a horizontal (Fig. 24.4a) and a vertical (Fig. 24.4b-d) single-shell steam generators with submerged heating surfaces and internal separators. The heating surfaces of these generators are made from tubes of a small outside diameter (12-22 mm) and a wall thickness of 1.2-1.5 mm. In high-capacity steam generators, the number of tubes may be as large as

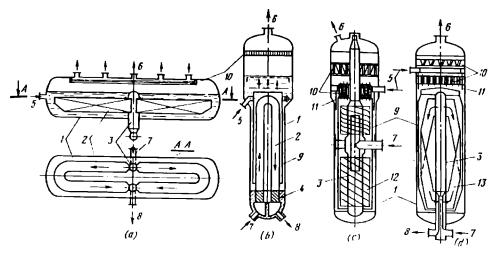


Fig. 24.4. Steam generators with aqueous coolant

(a) horizontal type; (b) vertical type with U-shaped tubes; (c) ditto, with helically colled tubes; (d) platentype; I—housing; 2—U-shaped tube system; 3—header and tube sheet; 4—tube plate; 5—food water in; 6—steam out; 7—coolant in; 8—coolant out; 9—jacket; 10—separator; 11—feed water distributing tubes; 12—bundle of helically colled tubes; 13—platens

some ten thousand. The ends of tubes are expanded in tube plates. High tightness of joints is quite important, since any leakage can transfer radioactivity from the primary to the secondary circuit. Since contamination of the aqueous coolant by corrosion products is intolerable, all elements of the primary circuit are made of stainless steel.

In order to minimize the use of metal in the generator housing, the heat-transfer agent having a higher temperature and pressure is circulated in tubes. The circuit containing the coolant operates under the conditions of forced circulation. For the working fluid (secondary circuit), natural circulation is preferable. The steam space of the drum is used to deliver steam with the least concentration of impurities. The design of steam-cleaning devices is determined by the position of the drum and the conditions of steam supply to the disengagement surface. In a horizontal steam generator (Fig. 24.4a), the coolant has a variable temperature along the length of steam-generating coils: it is the highest at their inlet and the lowest at the outlet. Therefore, the intensity of steam generation is also different in various portions of the drum. The intensity of evaporation from the disengagement surface is also different and is equalized by arranging a submerged perforated plate or by differentiated supply of feed water to the coils. Since the average intensity of evaporation from the disengagement surface in the steam generator is not high, steam of satisfactory quality ($\omega \leq 0.2\%$) can be produced by using steam-separating and steamwashing devices or only the former.

In vertical steam generators (Fig. 24.4b-d), the intensity of evaporation from the disengagement surface is much higher than in the horizontal type and results in a higher carry-over of moisture by steam. In order to diminish the moisture content of steam, the steam space of the drum has a substantial height so as to accommodate two or three steam-separating stages.

In vertical steam generators with U-shaped tubes (Fig. 24.4b) thermal expansion of the tube bundle is self-compensated; this is a serious advant-

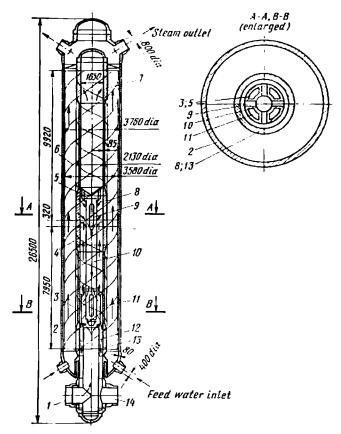


Fig. 24.5. Once-through steam generator for 500 MW 1-coolant in; 2-housing; 3-lower water distributor; 4-beader; 3-upper water distributor; 6-tube bank (heating surface); 7-upper tube plate; 8-middle tube plate; 9-vertical slite; 10-insert; 11-sector channels; 12-shaft; 13-lower tube plate; 14-coolant out

age. On the other hand, the tube plate should be of an appreciable thickness. In high-capacity steam generators, with the 4 000-mm diameter of the housing in the narrower section, the thickness of tube plate may reach 600-700 mm. The manufacture and assembly of such plates is complicated and expensive, since a large number of deep holes must be accurately drilled for expanding of tube ends. The water in steam generators may contain ironoxide sludge which is removed by blowing-down. In steam generators having a bottom tube plate, removal of sludge involves certain difficulties, since it accumulates in the bottom portion of the tube bundle just near

the tube plate. The sludge layer in this place is gradually evaporated; this results in increasing concentration of all water impurities (including chlorine ions and alkalies) in pores of the sludge layer, which can destroy even stainless-steel tubes.

In that respect, vertical-type steam generators without the bottom tube plate are advantageous and promising (Fig. 24.4c). It has two sections. The lower section accommodates the heating surfaces made from coiled stainless-steel tubes expanded in tube plates. The coolant (water) at a pressure of 16 MPa moves in the tubes. The inlet and outlet of the coolant are divided by partitions (in Fig. 24.4c,

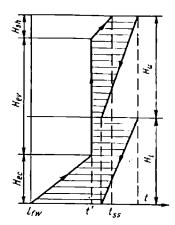


Fig. 24.6. Temperature distribution in coolant and working fluid and temperature gradients along the height of a once-through steam generator (see Fig. 24.5)

only the inlet is shown). Feed water is introduced from the top of the tube system. The latter is submerged into water from which saturated steam at a pressure of 6.4 MPa is produced. Water moves by gravity circulation. The jacket that surrounds the tube system forms a descending channel together with the housing. Inside the jacket, the fluid moves ascendingly. The upper section, which has a larger diameter, forms the steam space in which two stages of steam separators are arranged: the first stage (for coarse steam separation) has cyclone separators with axial supply of steam-water mixture; the second stage (for steam drying) has vertical annular louvertype separators. An advantage of steam generators with helically coiled tubes is that temperature expansion is selfcompensated. The principal drawback consists in complicated manufacture of helical coiling, since the angle of coiling is different in various tube rows.

Certain advantages are offered by steam generators with platen-type heating surfaces (Fig. 24.4d). Their platens are essentially a system of vertical tubes of the same length which are bent 180° midway of their height to provide proper rigidity. The platens

are radially arranged in vertical planes around the header. Headers in vertical-type steam generators (Fig. 24.4c and d) have a relatively small diameter and thin walls.

Figure 24.5 shows a 500-MW vertical once-through steam generator designed at the All-Union heat engineering institute. It has a vertical housing provided with pipe connections for the supply of feed water and removal of weakly superheated steam. A header is mounted coaxially inside the housing; it has pipe connections for the supply and removal of aqueous coolant. In the annular space between the housing and header a tubular heating surface is mounted which comprises two bundles of 12 mm \times \times 1.2 mm tubes 14 m long. The top and bottom tube bundles are connected in parallel for the coolant and in series for the working fluid. The latter moves in the intertubular space (Fig. 24.6).

24.3. Steam Generators with Liquid-metal and Gaseous Coolants

Steam generators with a liquid-metal coolant. Liquid metal passing through the reactor gets activated, so that one heat exchanger turns out to be insufficient and the heat-exchange system becomes more complicated. To make the steam generator safe in operation, the nuclear power station has three circuits and two successive heat exchangers (see Fig. 1.2c). In the first heat exchanger, the heat of liquid motal is transferred to an intermediate heat-transfer agent and in the second heat exchanger (the steam generator proper), the heat of the intermediate heat-transfer agent is utilized to produce steam. In plants with three circuits, sodium is used in the first (primary) and second (secondary) circuit and water in the third (ternary) circuit.

In steam generators with liquidmetal coolant, the processes of steam generation and superheating are usually organized in two separate apparatus which are connected in series. Steam reheating is carried out in a third apparatus. All of them may be of the shell-type or shell-and-tube type. The main reason for using the separated design is to avoid welding of tubes of various steel grades which are employed in the heating surfaces of the steam-generating and superheating sections.

In the Soviet-made liquid-metal steam generator operating with a fast-neutron reactor type BN-600, the processes of steam generation, superheating and reheating are carried out in three separate heat exchangers of the shell-and-tube type, which are called modules. Three modules are combined into a section. Eight sections form a once-through steam generator. The sections are connected in parallel for the coolant and working fluid. The power-producing unit comprises a BN-600 reactor and three steam generators.

The modules of the superheater and reheater are connected in parallel for the coolant, which ensures a high temperature gradient in both modules. Liquid sodium from these modules flows at a constant temperature into the evaporating module. Feed water in the evaporating module is heated to the saturation temperature, evaporated, and slightly superheated (by 12-15 deg C). The fluids move in counter-current flow.

The evaporating and superheating modules are essentially of the same design. A version of the superheater module is illustrated in Fig. 24.7. The heat-exchange surface is made from straight tubes which are bent sinusoidally in the lower portion for compensation of thermal expansion of the shell and tube system. The tubes are expanded in tube plates and arranged in them in the form of hexagonal rows. The working fluid (water and steam-water mixture in the evaporator and steam in the superheater and reheater) moves in the tubes and liquid sodium, in the

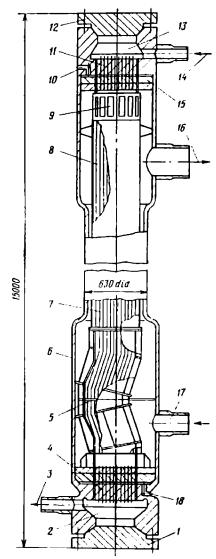


Fig. 24.7. Steam-generator section with liquid-metal coolant

1—bottom; 2—discharge chamber; 3—steam out; 4—lower tube plate; 5—spacer plate; 6—housing; 7—tube bundle; 8—jacket; 9—holes; 10—to gas reservoir; 11—upper tube plate; 12—cover; 13—inlet chamber; 14—steam in; 15—protective plate; 16—sodium out; 17—sodium in; 18—drainage

intertubular spaces. The shell is separated from the flow of liquid sodium by a jacket in order to equalize probable temperature variations of the coolant. The tube plates are

protected by displacing plates and insulating washers. Sodium is introduced into the tube bundle from an inlet chamber through 12 vertical slits 250 mm × 110 mm in the jacket and is thus distributed uniformly in the intertubular space. The chamber for sodium outlet is designed in a similar way. The inlet and outlet chambers for the working fluid aro formed by tube plates and detachable plane end covers of the shell.

The main parameters of the steam generator are as follows: heat power 490 MW, steam-generating capacity 181.6 kg/s, superheated steam pressure 14.2/2.45 MPa, superheated steam temperature 505°/505°C, coolant tem-

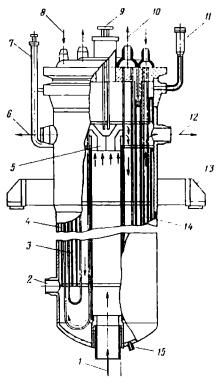


Fig. 24.8. Sodium-coolant steam generator of fast-neutron power reactor (Great Britain)

1—sodium in; 2—outlet of the products of interaction of sodium and water; 3—tube bundle; 4—thermal shield; 5—guide tubes; 6, 12—sodium out-released; 8—working fluid in; 9—level gauge at the sodium inlet side; 10—working fluid out; 11—hydrogen detector; 13—support; 14—lacket; 15—drainage

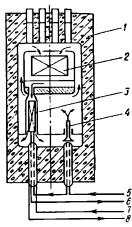


Fig. 24.9. Arrangement of steam generator elements in the housing of a power reactor 1-reinforced concrete housing; 2-reactor; 3-steam generator section; 4-gas blower; 5-steam to reheater; 6--reheated steam; 7-feed water; 8-superheated steam

perature: 520°C at the inlet and 320°C at the outlet.

A shell-type steam generator with sodium coolant is shown in Fig. 24.8. The outside diameter of the shell has been taken to be not more than 3 m for ease of manufacture and transport. The heating surfaces of all the heat exchangers are essentially of the same design: they are made in the form of U-shaped tubes arranged on the shell periphery. Since the pressure of liquid metal is lower than that of the working fluid, the former is caused to circulate in the intertubular space. A cylindrical jacket is provided in the shell in order to organize counter-current flow of the coolant and working fluid. A thermal shield prevents overheating of the shell.

The steam generator described has the following parameters: power 1 320 MW, superheated steam pressuro 16.3/3.4 MPa, superheated steam temperature 540°/540°C, the temperature of coolant: 560° at the inlet and 380°C at the outlet.

Gaseous-coolant steam generators. Modern steam generators with gaseous coolant are arranged together with the reactor in a common shell made of prestressed concrete (Fig. 24.9).

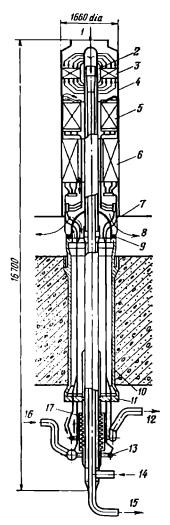


Fig. 24.10. Steam generator section of a nuclear power station at Fort St. Vrain (USA) (N=330 MW, D=1.040 t/h, $D_c=87 \text{ t/h}$)

1—coolant in; 2—labyrinth scaling; 3—reheater; 4—housing; 3—second-stage superheater; 6—economizer, evaporator and superheater of the first stage; 7—lower floor; 8—coolant out; 9—first cover; 10—section support; 11—second cover; 12—superheated steam out; 13—feed water header; 14—cold reheat steam; 15—hot reheat steam; 16—feed water in; 17—bellows

The steam generator comprises a number of parallel sections mounted circumferentially beneath the reactor core or in an annular spacing around the core. The heat-transfer agent between the steam generator sections is

moved by a gas blower driven by worked-off reheat steam. The steam generator is of the once-through type $(p = 17.3 \text{ MPa}, t_{ss} = 540^{\circ}\text{C} \text{ and } t_{rh} = 545^{\circ}\text{C})$. The gas has a temperature of 406°C at the reactor inlet and 780°C at the outlet and a pressure p = 4.7 MPa. The flow rate of gas is $1.6 \times 10^6 \text{ kg/h}$.

A section of the steam generator (Fig. 24.10) has three heating surfaces: the first of them includes the economizer, evaporator and the first superheater stage; the second is formed by the second stage of superheater, and the third comprises the reheater. The heating surfaces of the economizer and evaporator are finned on the gas side in order to intensify heat transfer. The coolant (helium) moves from the top downwards. Feed water is supplied into an annular header and is then distributed by tubes between intermediate headers. The tubes passing from a header form a heat-transfer surface (a bundle of helical coils). Water in this surface is heated to boiling, evaporated, and the steam is slightly superheated. The working fluid moves upwards in counter-current flow relative to the coolant.

Tubes from the top portion of the first bundle pass through an annular space formed between the shell of the section and the jacket of the secondstage superheater and descend in the form of helical coils in which the working fluid moves in parallel flow relative to the coolant. In the bottom portion of the superheater, tubes pass through an annular space between the supporting cylinder and the internal jacket of the lower tube bundle and are connected to intermediate headers below that bundle. Superheated steam is fed from the intermediate headers into a common annular header. The circulating tubes for superheated steam and feed water have helical loops for thermal expansion compensation. The counter-current reheater has helical tube coils connected to headers.

The mass of all tube bundles is supported by a spherical cover and

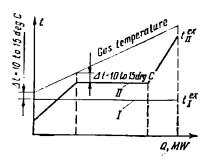


Fig. 24.11. t, Q-diagram of gaseous-coolant steam generator

transferred through a section support onto the reinforced-concrete housing. The space between the end covers of the apparatus is filled with helium at a pressure slightly above that in the reactor. The difference in thermal expansion of the shell and tubes of the reheater is compensated by a silphon.

Gaseous-coolant steam generators should preferably be made with an economizer and superheater. This is demonstrated in Fig. 24.11 where t_I^{ex} is the exit temperature of the working fluid when producing saturated steam and t_{II}^{ex} is the temperature of superheated steam at the same temperature of the gaseous coolant.

Indeed, for intensive heat transfer from the coolant to the working fluid, the temperature gradient Δt on the colder side should not be less than 10 deg C. When there is no economizer, generation of saturated steam, which is characterized by a constant temperature of the working fluid in the steam generator (line I), is limited to producing steam at a temperature t_I^{rx} corresponding to the given value of Δt .

An economizer permits of increasing the temperature gradient at the inlet, and therefore, raising the exit temperature of steam (line II). If a superheater is added to the economizer, a higher exit temperature $t_{II}^{\rm ex}$ can be obtained.

24.4. Nuclear Reactor as a Steam Generator

In single-circuit nuclear power stations, the functions of steam generator are performed by a boiling reactor in which steam is produced from the feed water being supplied. Thus, water serves as the coolant of the reactor and the working fluid from which steam is generated. Boiling reactors may be of shell or channel type.

Shell-type boiling reactor. A modern shell-type boiling reactor with forced circulation of water from which saturated steam for a turbine is produced is shown in Fig. 24.12. The reactor has a vertical cylindrical steel shell (drum) with a diameter of 3-6 m. The reactor core is arranged in the water space of the drum. For the reliable removal of heat from the core, water is intensively circulated by means of jet pumps mounted in the shell. In high-capacity power-producing reactors, the number of jet pumps may be up to 20. The pumps are arranged in the annular space between the shell and reactor core. Water behind the pumps is separated into two flows: the greater flow (2/3 of the total flow rate of circulating water)

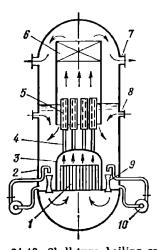


Fig. 24.12. Shell-type boiling reactor to core; 2—housing; 3—collecting chamber; 4—separator uptakes; 5—primary separators; 6—steam drier; 7—steam to turbine; 8 feed water supply; 9—jet pump; 10—circulation pump

passes through the core and the smaller flow is pumped by the circulation pumps through two parallel loops outside of the reactor shell. This portion of water is fed to the intake of the jet pumps and thus serves as the working fluid for the pumps. These external elements somewhat diminish the radiation safety of the reactor.

The cylindrical shell houses the reactor core with fuel assemblies supported by a plate. At the top, the reactor core has a spherical cover which forms the chamber for a steamwater mixture. Vertical tubes are connected to the spherical cover (one tube for each cell of the core). Steamwater mixture passes through these tubes into turbulent separators. Moisture separated in the turbulent separators is drained into the space between them and flows into the water space. The separated water is mixed with feed water and supplied to the intake of the jet pumps which overcome the resistance in the multiple forced circulation circuit. Separated steam is directed from the separators into the steam space where it is dried in steam driers to a moisture content roughly 0.1 %. Since the steamsoparating means are arranged inside the reactor shell, the circuit has no oxternal elements, which improves the radiation situation at the reactor.

Shell-type boiling reactors have been thoroughly developed and are used most widely. They are highly compact, have a simple circuit, low working pressure (roughly half the pressure of coolant in the primary circuit of water-cooled water-moderated power reactors), relatively low consumption of structural materials for the reactor core, and a low cost. Limited dimensions of the shell-type reactors are defined by the conditions of their transportation to construction sites. The manufacture of large-diameter shells may involve difficulties due to an appreciable increase of the wall thickness, especially in high-pressure plants. For this reason, the capacity

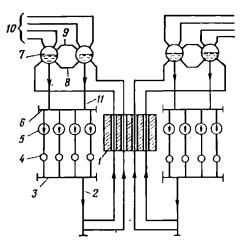


Fig. 24.13. Diagram of multiple forced circulation in channel-type boiling reactor type RBMK-1000

1—reactor; 2 distributing group header; 3—supply header; 4 return valve; 5—main circulation pump; 6 discharge header; 7—separating drue; 8 and 9 equalizing tubes; 10—steam to turbine; 11—downtake tubes

of shell-type reactors in power units is limited to 1 000 MW. The reactor shell, especially when it has large dimensions, operates under heavy conditions, since it is subjected to the action of high pressure of the coolant and of neutron flux in the core.

In shell-type reactors, it is difficult to collect information for predicting the appearance of defects in the reactor bousing, especially in highcapacity plants where a large volume of welding is done at the construction site, rather than at the manufacturing work. Shell-type reactors can operate with natural or forced circulation and produce steam of subcritical parameters.

A channel-type boiling reactor (Fig. 24.13) has a core which is a graphite block with vertical fuel channels provided in a definite order. Water in the channels moves around fuel assemblies, is heated to boiling and partially evaporated.

The adequate performance of boiling reactors at a high heat release rate had made it possible to develop

reactors in which steam is not only generated, but also superheated to the required temperature. A number of fuel channels is isolated for steam superheating. They are fed with saturated steam which has been separated from steam-water mixture in the reac-

When a channel-type reactor operates as a steam generator, its steamgenerating capacity is determined by the number of channels and the steamgenerating capacity of each channel. With a given steam-generating capacity of a channel, the total capacity can be increased by providing more channels in the core, but this complicates the design and operation of the plant.

Channel-type reactors can be built of a substantial unit capacity (1000 MW, 2000 MW and more) and can operate at high parameters of the coolant, and therefore, have a high efficiency.

The existing channel-type reactors operate with forced multiple circulation, but their design is suitable for once-through steam generation.

The advantages offered by channeltype reactors (the principal advantage being that the pressure in the shell is not high) have led to the development of nuclear power stations with batch produced boiling reactors of high capacity. The channel-type uranium-graphite reactor type RBMK--1000 has been designed for producing saturated steam. The unit operates by the single-circuit scheme, i.e. the steam generated in the reactor is directed immediately into the turbine and the turbine condensate is returned into the reactor circuit.

The reactor is mounted in a concrete well with a cylindrical stacking of vertical graphite columns; these have central borings (fuel channels) in which fuel elements are placed and coolant is circulated. Within the reactor core (its height is roughly 7 m), the fuel channels are made from 88 mm × × 4 mm zirconium tubes. Beyond the core, they are made of stainless steel. Each channel has a cassette with two fuel assemblies (with 18 fuel elements in each assembly). A fuel element consists of a zirconium tube (13.5 mm \times \times 0.9 mm) filled with uranium dioxide tablets. Some of the channels carry the absorption rods of the control and protection system.

The heat released in the reactor is removed by two loops of the multiple forced circulation circuit. The circulation of the coolant in each loop is effected by means of four main circulation pumps (with three pumps being in operation and one in reserve). Water after a pump (at a pressure around 8 MPa and temperature 265°C) is fed into a supply header and then into distributing headers and is distributed by means of control valves between the fuel channels. Water in the fuel channels movesfrom then bottom upwards and is first heated in the economizer section and then partially evaporated. The steam-water inixture thus formed (x = 14.5%) is directed through individual pipelines into separating drums. The two separating drums in each circulation loop are connected by a cross-over tube in order to equalize the water levels in them. The steam-water mixture is separated in the drums into steam and water. Saturated steam (with the moisture content 0.1%) is directed into turbines and water is mixed with the turbine condensate, is purified, preheated and deaerated, and returned by circulation pumps to the inlets of the fuel channels.

As a next step, the channel-type reactors for a capacity of 1500-2 000 MW with steam superheating in the reactor channels at a pressure of 6.5 MPa are now being developed.



METALS FOR STEAM BOILERS

25.1. Metal Behaviour at High Temperatures

We have discussed in earlier chapters some methods for organization of boiler processes which can form optimal conditions for the operation of the boiler element metal at high pressures. Even then, however, the metal of critical hoilor elements (waterwall tubes, drum, headers, steam pipelines) operates under heavy conditions. In steady regimes, it is subjected to stresses caused by internal pressure and the own mass of the elements and to a high temperature. In unsteady regimes (start-up, shutdown) it is subjected, in addition, to variable temperatures and pressures. Under certain conditions, cyclic variations of temperature may take place.

Besides, some critical elements at high temperatures can be acted upon by corrosive media: flue gases, saturated and superheated steam, steamwater mixture, and feed water, which can cause corrosion of the metal.

Many boiler elements, especially water-wall tubes and fittings in the water-steam path, operate under the conditions of erosion and abrasion wear. Erosion is the kind of wear produced by jets of liquid and abrasion is the mechanical wear under the effect of solid abrasive particles (for instance, abrasion wear of water-wall tubes by ash). Erosion of metal is a complicated process which combines chemical corrosion produced by an aggressive medium and mechanical wear as the dynamic effect of a flow of liquid or steam (mechanical wear in erosion can be enhanced by solid abrasive particles carried by the fluid).

In air heaters, the metal of their tubes is subjected to a relatively low pressure (up to 7-8 kPa), and therefore, mechanical stresses are insignificant, but the metal operates at a high temperature (up to 400-450°C).

The non-cooled fastening and suspending elements in boiler gas ducts operate under especially heavy conditions. They carry a high weight load and are subjected to the effect of aggressive gases at high temperatures (up to 800°C).

The boiler structure also carries a high weight load, but operates at the temperature of the surrounding air. Only few elements of the boiler structure of suspended design ('hot suspensions') operate at an elevated temperature determined by the temperature of the working fluid that flows through them.

All boiler elements oporating at elevated pressures can be divided into two groups:

(1) those which operate at temperatures below 350-400°C; they include the drum (in drum-type boilers), steam-generating tubes and their headers, tubes and headers of the economizer and transition zone, and pipelines and fittings for water and saturated steam; and

(2) those operating at temperatures above 350-400°C: steam superheaters and their headers, desuperheaters, superheated-steam pipelines and fittings.

Of special danger for the operation of critical boiler elements is the combined long effect of the internal pressure and high temperature (above 450°C) of superheated steam (superheater tubes and headers and steam mains). This may lead to creep

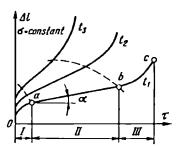


Fig. 25.1. Creep curves of stéel at various temperatures and constant stress

a special kind of plastic deformation in which the strain increases slowly and continuously at a constant stress below the yield stress and the size of an element gradually increases. As the residual plastic deformation attains a definite limit, rupture of the metal occurs. For this reason, creep is controlled by periodically measuring the dimensions of suspected elements.

Figure 25.1 shows creep curves obtained at a constant stress and different temperatures ($t_1 < t_2 < t_3$). Let us analyse the process of creep at temperature t_1 . The curve at t_1 can be divided into three portions: Oa, ab, and bc. Portion Oa (the period of attenuating creop) corresponds to a short initial period I when the metal is even strongthened slightly. Then follows a longer period II of steadystate (secondary) creep (portion ab) during which an element still can operate reliably without rupture. The rate of creep during that period is constant:

$$v = \Delta l/\tau = \tan \alpha \qquad (25.1)$$

In the final critical period III of accelerated creep (portion bc), a high plastic deformation occurs in an element (for instance, 'inflation' of tube portions in water walls), after which rupture takes place at point c. Reliable operation of elements is only possible within the period II of steady-state creep. At a higher temperature $(t_2 \text{ or } t_3)$, the process of creep occurs in a similar way, but more quickly;

the rate of steady-state creep increases and rupture takes place earlier. The stress at which the rate of creep during period II does not exceed the specified value or the stress that causes the total plastic deformation during a specified time of operation not above a safe limit, is called the creep stress, or creep strength σ_{cr} . For most steel grades, the allowable total plastic deformation of 1% after 100 000 h of operation is allowed, which corresponds to the creep rate $v_{cr} = 10^{-7}$ mm/(mm h) or 10^{-5} %/h.

The strength of the metal in operation under creep conditions can be characterized by the long-term strength. As the metal is stressed under creep conditions, the time of its reliable operation up to rupture depends on the stress applied. The stress which causes rupture of the metal in creep during a specified period is called the long-term strength.

The relationship between the stress and the time to rupture at constant temperature τ_p is most often expressed by an exponential equation:

$$\tau_n = B\sigma^{-m} \tag{25.2}$$

where B and m are constants for a given metal and given temperature.

In logarithmic coordinates, the relationship between σ and τ_n is described by a straight line; in Fig. 25.2, such lines are shown for three different temperatures t_1 , t_2 and t_3 ($t_1 < t_2 < t_3$).

Such curves can be used for determining the time of safe operation of an element (without rupture). There-

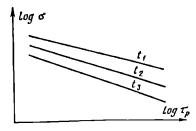


Fig. 25.2. Relationship between long-term strength and time to rupture

fore, the elements operating under the conditions of creep can be calculated by the long-term strength rather than by the creep strength. Giving the time of safe operation and a particular safe margin, we can calculate the stress at which these conditions will be satisfied.

The physical nature of the creep stress is different from that of the long-term strength. The creep strength (stress) characterizes the resistance of a metal to a low plastic deformation at an elevated temperature, while the long-term strength characterizes its resistance under the conditions of creep. Nonetheless, either characteristic can be used in creep calculations; moreover, there is a definite relation between these characteristics of a particular material.

Until recently, boiler elements operating under creep conditions were calculated by using in the formulae the allowable stress as found from the long-term strength at a given temperature of the wall for a specified period of operation (usually 100 000 h). The long-term strength is denoted σ_{tt}^t (where t is the wall temperature). With the operation factor $K_{op} = 0.85$, the operating time $\tau=100\,000$ h is roughly equivalent to 15 years. For expensive equipment, such as steam boilers and turbines in which the replacement of certain elements (steam pipelines, superheaters, headers, the high-pressure portion of the turbine, etc.) is labourconsuming and costly, this time of operation (campaign) is now considered to be too short in view of the high safety margins on which their strength has been calculated. It is now recommended to increase the depreciation time roughly twice, i.e. up to 200 000 hours. This can reduce somewhat the long-term strength at the end of the service life of the metal compared with that at the end of a period of 100 000 hours, but the reduction will not be high in view of the improved technology of manufacture and heat treatment of steels,

better maintenance of the equipment, and more perfect methods of metal control in the operating equipment. According to the new norms of strength calculations, elements of steam boilers can be calculated for a service life of 200 000 hours.

The operation of a metal at elevated temperatures can also be characterized by diffusion processes which can substantially change the structure and properties of the metal. Under such conditions, the metal can lose strength owing to embrittlement and graphitization. In some cases, these effects can lead to emergency situations.

The operation of steels under the effect of flue gases and superheated steam at elevated or high temperatures can also be characterized by the activization of electrochemical corrosion and, as a consequence, by the oxidation and scaling of surfaces in contact with the active media. Flue gases have an especially strong effect on the external surfaces of superheater tubes and superheated steam, on their internal surfaces, superheated steam headers, and steam main pipelines. In some cases scale formation may be so strong that the thickness of tube walls decreases to a dangerous value and may involve promature creep and even tube rupture. Scaling is aggravated by intensive heating loads or high stresses due to the pressure in tubes. Thinning of tubes due to scaling should be properly considered in strength calculations.

As the unit capacity of power plants increases, the requirements for their reliability become more rigorous and can be satisfied with greater difficulties in view of the large consumption of metal in the heating surfaces and enormous number of welded joints. An increase in unit capacity is often associated with an increase of steam parameters, which requires the application of steel grades having a higher strength and therefore, lower ductility. They are more sensitive to stress concentrations, so that even

slight defects of the metal become dangerous.

As a result, modern steam boilers of high capacities require a better technology of manufacture and careful metal testing which can provide ample information on the structure, composition, mechanical properties (including creep strength) and the stresses that may appear in the metal in operation.

For operation of steels at high temperatures and pressures, it is possible to formulate the following principal requirements which can ensure a long and reliable operation of boiler plants: the metal should possess high values of creep strength and long-term strength, high scaling resistance, stable structure which can ensure that its properties will not change dangerously in long-term operation, good weldability, and absence of metallurgical and mechanical defects on the surface, which might serve as stress concentrators and weaken the cross section of boiler elements.

25.2. Metals for Steam Boilers

The main materials for boilers are carbon steels and alloy steels; the latter can contain chromium, nickel, molybdenum, tungsten, vanadium, etc. Most of these alloying elements are expensive, but their addition in minor quantities imparts valuable properties to steel, which are unattainable in carbon steels.

Carbon (unalloyed) steels are used for making steam boiler elements which operate under no-creep conditions, i.e. at temperatures not higher than 450°C. As required by the technology of welding which is the principal process in boiler making, many critical elements of boilers are made of low-carbon steels (grades 10 and 20). Steel 20 is predominantly used since it has a higher strength than grade 10 and is not inferior to this in weldability and corrosion resistance. In the microstructure of steels for boiler

tubes, soft and plastic ferrite is the main component, while the concentration of the hardening component, pearlite, is not high. Sheet steel has an elevated content of carbon, in the average from 0.15% (grade 15C) to 0.25% (grade 22C) and has a higher strength and satisfactory weldability. Steel grade 22C is characterized by an elevated strength, which is due to a higher content of manganese and minor addition of titanium.

Low-alloy pearlitic-class steels. A steel is low-alloyed if it contains up to 4-5% of all alloying elements. Such steels are used for making boiler elements for operation under creep conditions: superheater tubes and headers, and steam pipelines. They are also employed for making boiler drums for pressures up to 18-18.5 MPa.

Low-alloy steels resistant to creep at temperatures up to 580°C, if a very high scale resistance is not required, are called heat-resisting, or high-temperature steels. Those which are creep-resisting at temperatures above 580°C and have a high scale resistance at these temperatures are called refractory steels.

The principal alloying elements in steels are Mo, Cr, Si, and Al. Molybdenum is dissolved in ferrite and increases the long-term strength and creep strength. Chromium, silicon, and aluminium increase scale resistance, since they form with oxygen dense refractory oxides Cr_2O_3 , SiO_2 and Al_2O_3 whose coefficients of expansion are very close to that of steel. These compounds protect a steel against oxidation.

Boiler making uses widely the lowalloyed Cr-Mo steel of pearlitic class (grade 15KhM with 1% Cr and 0.5% Mo) and Mo-Cr steel grade 12MKh (0.5% Cr and 0.5% Mo). These steels, especially 15KhM, are characterized by good weldability, elevated creep strength, and low liability to graphitization.

The trend for increasing the temperature of superheated steam with the use of inexpensive low-alloyed steels of pearlitic class has led to the appearance of Cr-Mo steel additionally alloyed with 0.2-0.3% vanadium. Vanadium is a strong carbideformer and thus is favourable for increasing the creep strength.

In modern boiler making, wide use is made of Cr-Mo steel 12Kh1MF (1% Cr, 0.3% Mo, 0.2% V) and steel 15Kh1MF (1% Cr, 1% Mo, 0.2% V) which has a slightly higher carbon content and substantially higher content of molybdenum and possesses a higher creep strength. The minor addition of vanadium decreases the rate of creep. These steels can operate at temperatures up to 565-570°C.

Pearlitic steel grade 12Kh2MFSR has a very high scale resistance and high refractoriness; its high scale resistance is obtained by adding 2% Cr and 0.4-0.7% Si. The high refractoriness is obtained by a very small addition of boron (0.003-0.005%). This steel is, however, very sensitive to heat treatment conditions. It is used for making superheater tubes.

High-alloyed austenitic-class steels. The trend to raise the temperature of superheating up to 600-650°C has led to the application of steel possessing even higher heat and scale resistance. The structure of these steels is mainly formed by high-alloyed Cr-Ni or Cr-Ni-Mn austenite. Austenitic steels have a high scale resistance due to a high concentration of chromium. In contrast to low-alloyed grades, high-alloyed austenitic steels may contain up to 30% of nickel and chromium or even more; their cost is, however, several times that of the former. Titanium and niobium (stabilizing elements) are added to austenitic steels in order to prevent intercrystalline corrosion. These elements are strong carbide-formers and can combine all the carbon into carbides, thus preventing the formation of chromium carbides at the boundaries of austentic grains. If, however, chromium carbides have formed in the structure, austenite around them is depleted in chromium; these low-chromium regions lose corrosion resistance which may lead to intercrystalline corrosion.

The capability of steels for forming a purely austenitic structure can be increased by increasing the Ni/Cr ratio. Among steels of this type, it mentioning is worth steel grade 12Kh18N12T. and also grade Kh14N14V2M (which contains tungsten and molybdenum) and steel type 16-13-3 (16% Cr. 13% Ni, and 3% Mo). Molybdenum and tungsten are added to austenitic steels in order to further increase the heat resistance, since they form high-disperse strong compounds Fe₂Mo and Fe₂W in the steel structure.

High-alloyed steels of martensitic and martensite-ferritic class. A draw back of austenitic steels is their lia. bility to cracking under the combined action of stresses and corrosive media (corrosion cracking) and formation of annular cracks in welds owing to a sharp drop of ductility in some portions of the weld metal on heating. Austenitic steels are expensive due to a high content of nickel. Attempts to lower the cost of heat-resisting steels and eliminate the drawbacks typical of austenitic steels have resulted in the development of no-nickel steels on the basis of 11-13% Cr with the additions of molybdenu**m, tungste**n and vanadium for higher heat resistance. With such a combination of the alloying elements, the structure of the steel is low-carbon martensite or martensite-ferrite, which determines the name of this class of steels.

Low-alloyed steels for operation under no-creep conditions. In boiler making, wide use is made of low-alloyed steels for operation at relatively low temperatures when the phenomenon of creep does not appear. The steel has a higher strength than carbon grades and is employed in order to decrease the wall thickness of boiler elements and thus reduce metal consumption. For instance, boiler drums for high steam parameters are made of Mn-Ni-Mo steel grade

Table 25.1. Principal Characteristics of Boiler Steels

		Mechanical	l proper temperat	tics at ure	room	
Grade	Alloying elements, %	Ultimate strength, MPa	Yield strength, MPa	Per cent elongation	Impact touglaness, kN,m	Applications
	Low-carbon	(unalloy	ed) pea	rlitic-	class st	eels
15K 20K	=	352 402	215 245	25 23	685 590	Drums and vessels for pressures up to 6 MPa and temperatures up to 450°C
22K		500	245	22	835	Drums for pressures of 6- 12.5 MPa
10 20	-	335 402	205 205	31 24	490	Vessels and pipelines for temperature of fluid up to 450°C; tubes of heating surfaces of steel 20 for temperatures up to 480- 500°C
	Low-a	lloyed pe	arlitic-	class :	stecls	
16GNMA		500	335	16	685	Boiler drums for pressures 14-18.5 MPa
15 G S	Mn = 0.9-1.3 Si = 0.7-1.0	470-590	343	16	490	Feed-water pipelines for pressures up to 38 MPa and $t_{10} \leq 450^{\circ}\text{C}$
12MKh	Cr = 0.4-0.6	412	205	22	685	Headers and pipelines for
15KhM	Mo = 0.4-0.6 Cr = 0.8-1.1 Mo = 0.4-0.6	430	215	21	59 0	steam temperature up to 475-540°C; from grade 12MKh, up to 540°C, from 15KhM, up to 540°C; suporheater tubes at two up to 540-560°C
12Kb1MF	Cr = 0.9-1.2 Mo = 0.25-0.35 V = 0.15-0.3	440	225	21	590	Boiler tubes and steam pipelines at two up to 580°C
15Kb1M1F	Cr = 1.1-1.4 Mo = 0.9-1.2 V = 0.2-0.3	577	343	16	490	Superheaters and steam pipelines at t_{to} up to 580°C
12Kh2MFSR	Cr = 1.6-1.9 Mo = 0.5-0.7 V = 0.2-0.35 Si = 0.4-0.7 B = 0.005	470	235	21	590	Superheaters at two up to 580°C
12Kh2MFB	Cr = 2.1-2.6 Mo = 0.5-0.7 V = 0.2-0.35 Nb = 0.5-0.8	392	-	30	_	Superheaters at two up to 560°C

Continued

	Alloying elements, %	Mechanical properties at room temperature				
Grade		Ultimate strength, Mra	Yield strength,	Per cent	Impact toughness. kN/m	Applications
	II igh-	alloyed a	ustenit	ic-class	steels	
Kh14N14V2M	Cr = 13-15 Ni = 13-15 W = 2-2.75	540	215	35	-	Superheaters and pipelines at t_w up to 650°C
12Kh18N12T	Mo = 0.45-0.6 Cr = 17-19 Ni = 11-13 Ti = 0.65	540	215	35		
Kh16N9M2	Cr = 15.5-17 Ni = 8.5-10 Mo = 1.5-2	590	245	7 0	2 940	Steam pipelines at steam temperature up to 650°C
09Kh14N19V2BR	Cr = 23-15 Ni : 18-20 W 2-2-3 Nb = 0.9-1.3 B = 0.005	490	215	38	1 380	Steam pipelines at steam temperature up to 700°C; superheater tubes at t _{in} up to 700°C
Kh16N16V2MBR	Cr = 15-17 Ni = 15-17 W = 2-3 Mo = 0.4-0.9 Nb = 0.6-1.0	510	215	38	1 380	Steam pipelines and su- perhoaters at tw up to 700°C
Kh16N14V2BR	B = 0.005 Cr = 15-17 Ni = 13-15 W = 2-2.75 Nb = 0.9-1.3 B = 0.005	520	235	38	1 380	

16GNMA (1% Mn, 1.2% Ni, and 0.5% Mo). Feed water pipelines for supercritical pressures are made of Mn-Si steel grade 15GS (1.1% Mn, 0.8% Si).

The principal characteristics of steels employed most widely for making tubes, drums, headers and pipelines of boilers are given in Table 25.1.

Cast iron (grey and scale-resisting) is also employed widely in boiler making. Grey cast iron (SCh) has high casting properties and is used for making various furnace devices: man- and view-holes, explosion valves, fittings and fastenings of boiler setting. It can operate at temperatures

up to 250-350°C. Scale-resisting cast iron (OCh) is alloyed with some elements (such as silicon) which endow it with scale-resisting properties. It is employed in spacing teeth of superheaters, tube suspensions and other fastening elements operating in high-temperature zones.

25.3. Strength Calculations

The calculation of strength is based on estimating the strength of the metal in terms of the ultimate load (stress) at the rated pressure of the working fluid, which makes it possible to consider the operating conditions of the metal more completely and carefully. The rated pressure is the pressure at the plant outlet plus the hydraulic pressure loss at the nominal load of the path from the discharge header of the superheater to the point where the element being calculated is located, $\Sigma \Delta p$; filling the elements by water or steam-water mixture is taken into account by the hydrostatic pressure of a column of fluid above the element, Δp_h . Thus, the design pressure is:

$$p = p_{ss} + \Sigma \Delta p + \Delta p_h \quad (25.3)$$

The hydraulic resistances and the pressure of the column are considered if their sum exceeds the pressure behind the superheater by at least 3%. An additional term appears due to the force of gravity acting on the element being calculated and other elements connected to it. The additional load is restricted by extreme values at which the safety margin diminishes within tolerable limits compared with the strength determined in terms of pressure.

The design temperature depends on the conditions of heating and cooling of an element. For unheated elements, the design temperature is taken to be equal to the temperature of the working fluid; for boiler drums, it is taken as the temperature of saturation at the pressure in the drum; for headers and connecting pipelines as the temperature of the working fluid in them.

For heated elements, the design temperature is found from formula (10.10) which accounts for the nature of thermal action on a heating surface. It is essential to analyse various portions of a tube bank, in particular those which have the highest temperature of steam and those with the highest heating intensity. It should also be considered that some tubes or a group of tubes may operate with maldistribution of heat Δt_{md} , i.e. the fluid temperature in them may exceed the average design temperature. This difference can be found from the calcu-

lation of the plant or determined by tests. For each particular case, the formula can be simplified and the design temperature is determined as recommended in [22].

Allowable stress. The allowable nominal stress is the stress used in calculations of the least thickness of walls and the highest allowable pressure for a particular metal and expected conditions of operation.

The allowable nominal stress is selected for one of the strength characteristics of the metal: the ultimate tensile strength at 20°C, σ_t^{20} ; the yield limit at 20°C and at the design temperature t, $\sigma_{0.2}^{20}$ and $\sigma_{0.2}^{t}$; the long-term strength at the design temperature (10⁵ hours to rupture), $\sigma_{t,1105}^{t}$; and the creep strength (to cause 1% deformation in 10⁵ hours) at the design temperature $\sigma_{cr1\%}^{t}$. The allowable nominal stress is taken as the lowest of these characteristics divided by the safety factor adopted.

Most elements of steam boilers and steam generators operating at a particular pressure of the working fluid are of cylindrical shape (drums, headers and tubes of the heating surfaces of steam boilers, shells, chambers and tubes of the heat exchangers of steam generators at nuclear power stations). Cylindrical elements subjected to an internal pressure can be calculated by the general formulae:

$$S = \frac{\rho D_{ln}}{2\Phi\sigma_{nl} - P} + C \qquad (25.4)$$

0

$$S = \frac{pD_{ex}}{2\varphi\sigma_{al} + p} + C \qquad (25.5)$$

where S is the cylindrical wall thickness, m, p is the design pressure, MPa, D_{in} and D_{ex} are respectively the internal and external diameter of an element, m, ϕ is the coefficient of strength of the element which might be weakened by a longitudinal weld seam or by holes for the expansion of tube ends, and C is an addition to the wall thickness which is introduced to account for probable variations of wall thickness in manufactu-

red tubes, thinning of the material in bends, and thinning from the formation of scale during the element service life.

Calculation of a drum. The drum is one of the most intensively stressed elements in boilers. The following kinds of stress can appear in a drum: from the internal pressure at the working parameters of the fluid; thermal stresses, which include stresses due to the temperature difference across the drum wall and those due to the temperature difference between the top and bottom portions of the drum. Thermal stresses across the wall may appear during heating or cooling of the drum. Those between the top and bottom portions of the drum also appear during start-up or shut-down of the boiler; during startup, for instance, the upper half of the drum, which is in contact with the condensing steam, is heated up more quickly than the lower half where the circulation of water is still too weak. At shut-down, on the contrary, the upper half is cooled more slowly than the lower, since the heat transfer in the steam space is one or two orders of magnitude smaller than in the water space.

In addition to the stresses indicated, a drum is subjected to the action of the mass of its metal and water and to the stresses caused by deformation of the attached water- and steam-circulating tubes and steam pipelines. These stresses are usually low and can be neglected.

In the regimes of accelerated starting-up or emergency shut-down (for instance, on rupture of tubes in the circulation circuit), thermal stresses may exceed the allowable values. Substantial stress concentrations may occur at the sharp edges of holes in the places of tube connection, especially of downtake tubes. During start-up and shut-down, these stresses vary from zero to the maximum, while in steady regimes they vary less substantially, but with a higher frequency in accordance with the temperature

pulsations on the internal surface of the drum, say, on variations of water temperature behind the economizer.

For the reliable operation of a drum in various regimes, the strength condition should be satisfied for any moment of operation, i.e. the resolved stress (which is found by considering all the factors mentioned) should not exceed the allowable stress of the metal at a given temperature. Thus, the strength calculation of a drum should consider all the stresses mentioned and the non-cylindricity of the drum and its solution is quite complicated. In the existing norms for the strength calculations of boiler elements, only the stresses from the internal pressure in a drum are considered. In order to limit thermal stresses, it is not advisable to allow the temperature differences between the adjacent elements of a boiler to be more than 50-55 deg C.

Headers are calculated essentially in the same way as boiler drums. In that case, one should consider the loss of strength due to holes for the connection of tubes. Headers have no longitudinal weld seams. A header has two transverse weld seams for the connection of end covers, but their effect on the strength is not considered.

Calculation of tubes in the heating surfaces and pipelines is carried out by the formulae given earlier, with $\varphi=1$, since seamless tubes have no weakened points. Tubes of heat exchangers subjected to pressure from both sides are calculated for the higher pressure, rather than for the pressure difference. If the higher pressure is in a tube, the calculation is done by formulae (25.4) and (25.5). With a higher pressure on the outside, a tube is compressed and is calculated by the formula

$$S = D_{ex} \sqrt{\frac{pm}{2.2E}} + C$$
 (25.6)

where E is the modulus of elasticity, MPa, and m is the safety (stability) factor; $m \approx 6$ for carbon-steel tubes and $m \approx 5$ for alloy-steel tubes.

25.4. Metal Control in Operation

As given earlier, the properties of steels are impaired at high temperatures. For that reason, it is essential to periodically control the state of the metal operating under the conditions of creep (at temperatures above 450°C). Periodic control of the metal is required for pipelines, headers, tube coils of superheaters, flanges, and other boiler elements. All points where measurements are carried out should have reliable thermal insulation of the same conductivity as in other portions of a pipeline or a header.

In order to control creep and the structure and mechanical properties of the metal in steam mains, they are provided with control sections 4.5 m long in easily accessible places near the boiler (see Fig. 25.3). If a pipeline is made of pearlitic or ferriteaustenitic steel, bosses are fastened to the surface of the control section so as to control creep by variations of their diameter. With tubes made of austenitic steels, bosses are not used. and the residual deformation is measured directly on the tube section by means of micrometers. Once in three years, a specimen is cut off from a control section for making mechanical and metallographic tests and chemical and carbide analysis of the main metal and weld seams (Fig. 25.4).

Bosses are welded on the surface of the control sections and, in addition. in straight portions of a pipeline more than 1.5 m long, in four points at two mutually perpendicular diameters. All measurements are made upon cooling of the metal at least to 50°C so as to exclude the effect of temperature. Creep in headers is controlled in a similar way. Creep in superheater coils is controlled periodically by means of measuring brackets; this makes it possible to prevent emergency situations.

The method of control sections is disadvantageous in that cutting of specimens is labour-consuming and the cut-off portions of tubes must be filled with metal by welding. The principal drawback, however, is that the variations in the metal properties in a control section cannot be fully representative of the variations in all other tubes, so that control of all tubes is required.

Methods have been developed for the control of all tubes by non-destructive testing, i.e. without cutting of specimens. Microstructural analysis of the metal in operating tubes is carried out in place by means of a portable microscope with a photographic camera, which is fastened to the tube to be tested. The surface of the tube should be prepared, i.e. ground and polished by means of a drill with a set of grinding and polishing discs.

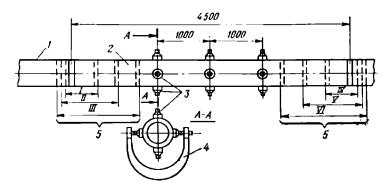


Fig. 25.3. Test section of a steam pipeline 1—steam pipeline; 2—test section; 3—bosses; 4—snap gauge for measuring residual deformation, 3—cutting specimens for tests; I-VI—cutting sequence

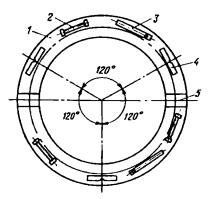


Fig. 25.4. Diagram of specimen cutting 1-cut-off portion of pipeline; 2-specimen for tensile tests; 3-specimen for impact toughness test; 4-specimen for carbide analysis; 5-specimen for making microsoction

Non-destructive mechanical tests are finding ever wider use for metal control at power stations. They are based on a definite relationship that exists between the hardness of a metal, as measured by indentation, and its principal mechanical properties.

In these methods, the strength characteristics of a metal, i.e. the yield strength $\sigma_{0.2}$ and the ultimate tensile strength σ_{t} are determined respectively by the hardness at the yield limit, $H_{0.2}$ and the Brinell hardness HB. The value of $H_{0.2}$ is found as the ratio of the indentation force $P_{0.2}$ (at an indentation load 2 450 N) to the surface area M of the indentation obtained on attaining a residual defor-

mation of 0.2%. With an indenting ball of 10 mm diameter, the residual deformation of 0.2% is obtained when the diameter of indentation is 0.9 mm and $M=0.6~\rm mm^2$. The ultimate strength σ_t is determined by measuring the Brinell hardness on indenting a ball 2.5 mm in diameter by the force 1840 N.

Ductility of a metal is characterized by the ductility characteristics: per cent elongation of a specimen with l/d=5 on tension, $\delta_{5\%}$, and its contraction ψ . Per cent elongation is determined by the formula

$$\delta_5 = \frac{A}{2\sigma_t + \sigma_{0.2}} 100 \qquad (25.7)$$

where A is the area under the stress-strain curve of a specimen tensioned to rupture, cm^2 .

Contraction ψ is found indirectly from the Brinell hardness of the metal.

Non-destructive methods of metal control are quite convenient, but their accuracy is somewhat lower than that of direct tests of metal specimens.

Thermal expansions of steam pipelines which usually form a complicated system in space, can cause redistribution of loads onto supports and result in excessive stresses in some portions of a pipeline. Expansions of pipelines operating at temperatures above 300°C are controlled by means of strain gauges.

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